# THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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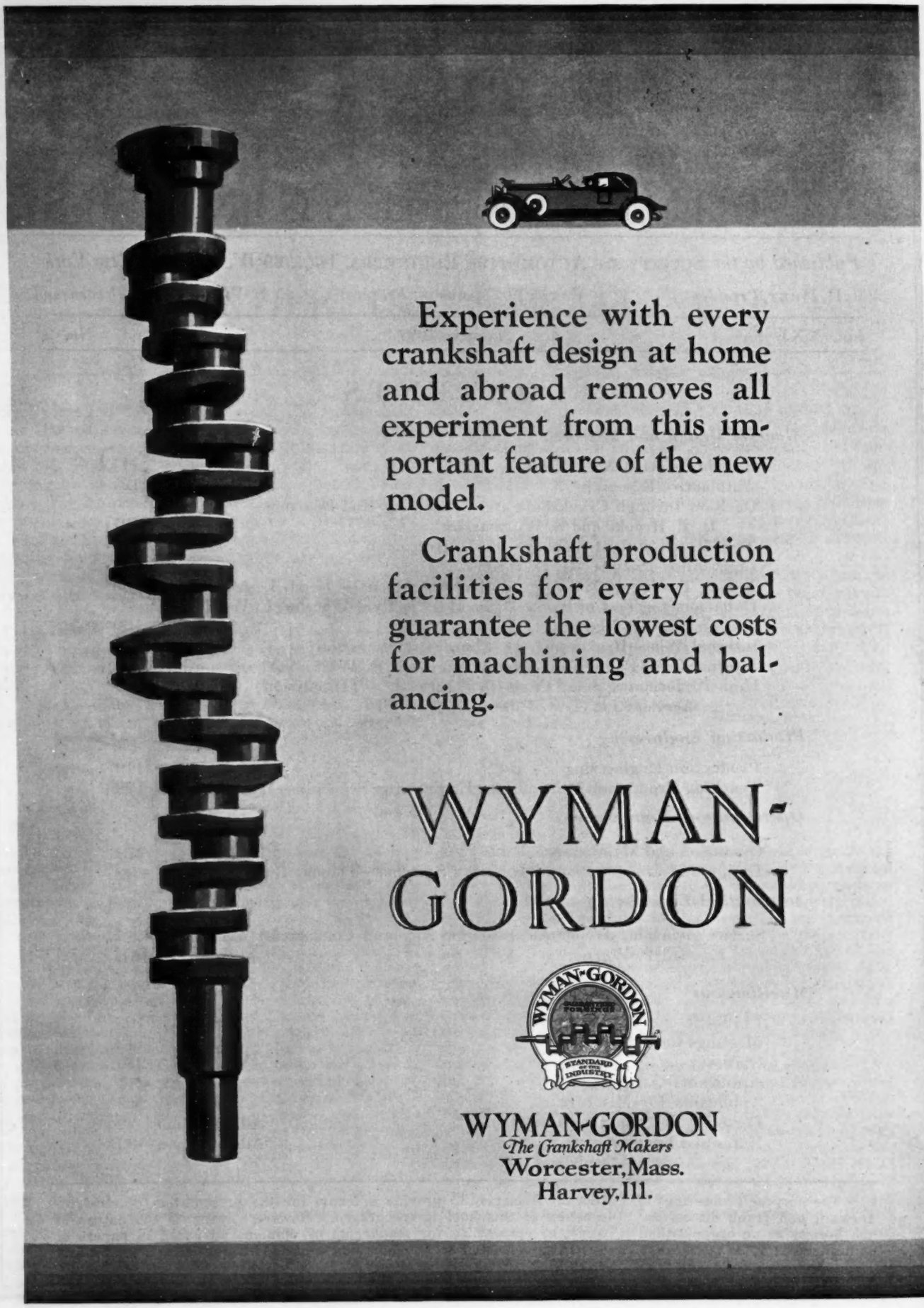
The purpose of meetings of the Society is largely to provide a forum for the presentation of straightforward and frank discussion. Discussion of this kind is encouraged. However, owing to the nature of the Society as an organization, it cannot be responsible for statements or opinions advanced in papers or in discussions at its meetings. The Constitution of the Society has long contained a provision to this effect.


Experience with every crankshaft design at home and abroad removes all experiment from this important feature of the new model.
Crankshaft production facilities for every need guarantee the lowest costs for machining and balancing.
WYMAN-GORDON
WYMAN-GORDON
The Crankshaft Makers
Worcester, Mass.
Harvey, Ill.

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XXI August, 1927 No. 2

# Chronicle and Comment

## Meetings Calendar

BEGINNING with this issue of THE JOURNAL, a calendar of National and Section meetings definitely scheduled will appear on the first left-hand page following Chronicle and Comment.

## Constitutional Amendment Adopted

A REPORT of the Tellers of Election shows that the proposed amendment to the Constitution providing for the composition and election of the annual Nominating Committee has been adopted. The amendment was set forth in the letter-ballot dated June 3, 1927. Of a total of 754 votes cast, 706 favored adoption of the amendment, 13 were opposed to it and 35 votes were defective. The tellers were S. R. Dresser, F. K. Glynn and C. B. Veal. As voted by the letter-ballot, the amended paragraph now in effect reads:

NOMINATING COMMITTEE

C 46 The annual Nominating Committee of the Society shall consist of one Member of the Society to be elected from and by each Section of the Society prior to the Annual Meeting; and three Members of the Society who shall be elected at the business session of the Annual Meeting preceding the Annual Meeting at which officers are to be elected; no two of said three Members shall reside in the same Section District.

## Changes in Section Officers

OWING to changes in company connection and assignment of Section officers, necessitating their moving from one city to another, several changes in Section officer-personnel have been made.

R. E. Plimpton, recently elected Chairman of the Metropolitan Section, has resigned owing to his moving to Chicago. E. W. Lowe, elected as Vice-Chairman, has been named by the Governing Committee to fill the vacancy, Sidney R. Dresser becoming Vice-Chairman.

F. G. Shoemaker, vice-chairman of the Dayton Section, has resigned because of his moving to Buffalo. The Governing Committee has not yet elected his successor.

Ludwig Majneri, treasurer of the Buffalo Section, has resigned on account of changing his residence to Detroit, Gustaf Carvelli succeeding him.

R. N. DuBois, who was elected Secretary of the Washington Section, now lives in Flint, Mich. The Governing Committee of the Washington Section appointed as Secretary E. S. Pardoe, who had resigned as Treasurer to make this transfer possible. G. E. Reynolds was selected to serve as Treasurer.

Mention should also be made of Norman G. Shidle, who was elected Chairman of the Pennsylvania Section. A. K. Brumbaugh had been nominated for Chairman, but moved to Cleveland before the time of the election. To overcome the difficulty in parliamentary procedure involved, Mr. Brumbaugh organized an additional Nominating Committee with himself as a member. This Committee nominated Norman G. Shidle for Chairman and campaigned successfully for his election.

## Charles L. Lawrance

TO few men comes the signal honor of accomplishing so superb a feat as that achieved by Charles L. Lawrance in designing the engine that carried Lindbergh, Chamberlin, Byrd, Maitland, and Smith in their successful transoceanic flights. Mr. Lawrance, president of the Wright Aeronautical Corporation, was largely responsible for undertaking, in 1920, the development of the air-cooled stationary radial engine that has been brought, during the last 7 years, to its present state of consummate excellence.

Mr. Lawrance was elected to membership in the Society in 1905 and has been prominent in its aeronautic activities. At the 1922 Annual Meeting his paper on Air-Cooled Engine Development was one of the most interesting and valuable contributions to aeronautic engineering.

## Economy in Motor-Vehicle Operation

COMMENTING upon the impression given from time to time that motor-vehicle operation is well developed economically, Chairman Plimpton, of the Operation and Maintenance Committee of the Society, expressed the view that every five-passenger automobile traveling on the road with one passenger is unduly expensive; and that the same is true of every motor-truck that stands idle much of the time or does not carry a goodly part of the load which it is capable of carrying when it is running. Great volume in goods carried does not, of course, mean efficiency. In various cases large fleets of motor-trucks are operated under inferior management, whereas obviously the magnitude of the operations involved warrant the application of the abilities of high-grade engineers and executives. Many efficient fleet-operators believe that, regardless of possible im-

provement in vehicles, only a beginning has been made in establishing sound and advanced operating methods.

### The 1928 Summer Meeting

As it is necessary for the hotel arrangements for Summer Meetings to be made far in advance, the Meetings Committee is now considering the location and time for the 1928 Summer Meeting.

Inasmuch as the last two meetings have been held at French Lick Springs, Ind., it is argued that it is necessary, in fairness to the Eastern members, that the 1928 Summer Meeting be held in the East. Possible locations are Spring Lake, N. J., the place of the 1923 and 1924 Summer Meetings; Quebec; Swampscott, Mass.; and Lake Placid, N. Y. Suggestions regarding the desirability of these or other places would be greatly appreciated by the Meetings Committee.

Although many members would appreciate having another meeting held at White Sulphur Springs, W. Va., where the 1922 and the 1925 meetings were held, the hotel facilities there are not adequate for the large number of members now attending Summer Meetings of the Society.

### Coming National Meetings

THE technical program for the Production Meeting, Sept. 19 to 22 inclusive, is rapidly taking final form and definite announcements covering it will be issued to the membership at an early date. The Stag Carnival, which has become one of the permanent features of Production Meetings, will be held at the Hotel Winton on Tuesday evening, Sept. 21. L. L. Williams is chairman of the Carnival Committee and an excellent dinner and an interesting show have been planned.

The Aeronautic Meeting this year, like the Production Meeting, will be a "two-city" meeting, sessions being planned to be held in connection with the National Air Races at Spokane, Wash., as well as in Chicago. The sessions in Chicago, which will constitute the principal part of the meeting, will be held probably during the week of Sept. 12, to afford the members ready opportunity to attend the dinner to the transoceanic fliers that has been proposed by the National Aeronautical Chamber of Commerce. This will also make it convenient for the aeronautic engineers who will go to the Spokane races to stop over for the meeting. Hon. Edward P. Warner is chairman of the subcommittee in charge of the Aeronautic Meeting.

Members of the Society will shortly receive official invitations to attend the Ninth Annual Meeting of the Army Ordnance Association on Oct. 6 at Aberdeen Proving Ground, Maryland. It is hoped that a large number of Society members will participate on this occasion, as the program will be well worth the attention of every member of the Society.

The Transportation Meeting will be held in Chicago in October. F. J. Scarr is chairman of the committee in charge of this meeting and the exceptional features that have been planned will be announced in an early issue of the *Meetings Bulletin*.

The Tractor Meeting will be held in Chicago on Dec. 1. O. W. Young is chairman of the committee in charge.

As reported in the article on the World Motor-Transport Congress on p. 115 of this issue, several members of the Society are seriously considering joining the European party that will visit the Congress and the Commercial Motor-Transport Exhibition. A formal request that the Society be represented at the Congress has been received from Horace Wyatt, organizer of the 1927 World Motor-Transport Congress. Copies of the announcement will be sent to members interested in making the European trip.

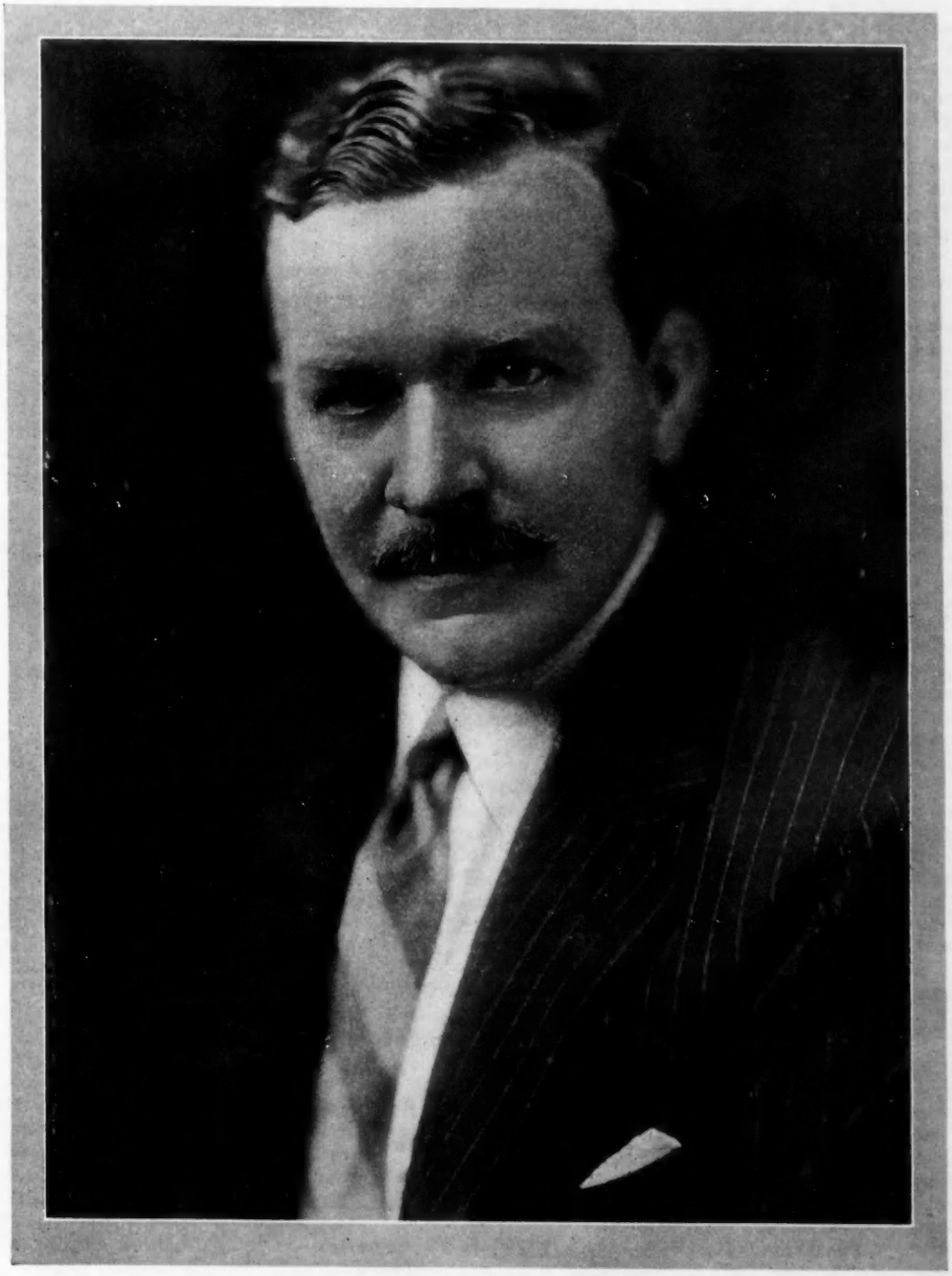
### International Standards

IN the matter of International Standards there are, of course, various views as to what the nature of these should be and to what extent they are possible and would be valuable in the automotive engineering field. Obviously, many of the Society's standards are of marked benefit to motor-vehicle users throughout the world. On principle, there should be standards making possible interchangeability of various units of automotive apparatus that are practically in universal use. This is dictated by the need for promptness and economy in servicing.

The Society has interchanged information with the Institution of Automobile Engineers of Great Britain in matters relating to standardization for many years. Considerable work has been done by the Society through Sectional Committees, organized under the procedure of the American Engineering Standards Committee, looking toward international agreement in standardization of ball bearings. There is still some question, however, as to how much can be accomplished in this direction. Conferences on the subject are being held currently in Europe, and it has been urged that this Country be represented at a meeting to be held at Stockholm, Sweden, in October.

At this time there is no existing organization in which the Society is participating formally that can declare anything to be an International Standard. In a general way at least the Society has accepted the American Engineering Standards Committee as the channel in this Country for international exchange of information on and cooperation in engineering standardization. This is analogous to the acceptance of the American Engineering Standards Committee as the National coordinating body of this Country under whose rules work should proceed that relates to more than one major industry, one of these industries being the automotive.

There has been much discussion recently regarding the organizing of a National committee or body in this Country that would have jurisdiction of all international matters affecting this Country in relation to standards and technology. Past-President Manly is a member of the American Engineering Standards Committee's Committee on Cooperation with Foreign Bodies, of which George K. Burgess, director of the Bureau of Standards, is chairman. While various organizations, including the United States Committee of the International Electrotechnical Commission, are engaged to some extent in international standardization work, it is felt that generally speaking and for certain purposes at least there should be a single organization having for this Country the inclusive jurisdiction mentioned above. An attempt is being made to settle upon a plan for such a body. The purpose is to secure the cooperation of all technical groups in coordinating properly their international relations in technical matters.



CHARLES L. LAWRANCE

# Meetings Calendar

## Production Meeting

Sept. 19 to 22, 1927

Cleveland Sessions
Hotel Winton, Sept. 19 and 20, 1927

Detroit Sessions
Hotel Statler, Sept. 21 and 22, 1927

## Aeronautic Meeting

Chicago Sessions September, 1927
Spokane Sessions September, 1927

## Visit to Aberdeen Proving Ground

Oct. 6, 1927

## Transportation Meeting

October, 1927
Hotel Sherman, Chicago

## European Trip

Nov. 8 to Dec. 3, 1927

World Motor-Transport Congress, London, England
Nov. 14 to 16, 1927

Commercial Motor-Transport Exhibition, London, England
Nov. 17 to 26, 1927

## Tractor Meeting

Dec. 1, 1927
Hotel Sherman, Chicago

## Joint Petroleum-Meeting

December, 1927, Detroit

A joint symposium of the Petroleum Division and the Division of Industrial and Engineering Chemistry of the American Chemical Society and the Society of Automotive Engineers on Sulphur in Motor Fuels.

# Operation and Maintenance

## THE S.A.E. ACCOUNTING STUDY

### Purpose of the Operation and Maintenance Committee's Accounting Subcommittee

In the early part of this year a Subcommittee on Accounting was organized under the Society's Motor Vehicle Fleet Operation and Maintenance Committee to make a thorough study of the accounting classifications now used by various classes of operators of large motor-vehicle fleets and to formulate a standard that would serve at least as the foundation for accounting classifications to be used by all large operators. One of the objectives of the work is to make it possible for operators to compare cost data directly and with the knowledge that they represent true conditions. R. E. Plimpton, chairman of the Operation and Maintenance Committee and also of the Subcommittee, has prepared the following paper after making a careful study of the present situation and analyzing systems used by various representative fleet operators. It is hoped that all such operators who wish to participate in this work will do so through the Society's office.

#### Present Vagueness of Fundamental Terms

Recent years have seen a tremendous increase in the number of motor-vehicles used for business or commercial purposes; we have the motorcoach, the taxicab and the truck in its manifold applications, yet there has been practically no advance in the general methods or practices of keeping a record of expenses incurred in their operation.

The real trouble seems to be that no two of us talk the same language. It is dangerous to refer to such terms as "acounts" and "cost keeping", because even these important terms have different meanings for different people. John Smith, garage superintendent for the Black Coal Co., is a big man in his organization. He has complete charge of 50 trucks, let us say; supervision over the drivers and hire-and-fire control of all shop and garage employes. Yet if you ask him whether his department keeps "accounts," 9 times out of 10 he will say "No." Try again, but say "costs"; in answering "Yes" he will appear surprised that you even ask such a question. Then he may explain that all the accounts are kept downtown in the main office, and that he knows very little about them.

Take a larger installation, where Edward Johnson has much the same responsibilities as Smith, but with the title of superintendent of transportation. Ask Johnson, who controls 500 vehicles for the Amber Oil Co., the same question about "accounts." He will reply promptly: "Why, of course. We are under the supervision of the company comptroller, but all accounting, as well as cost keeping for transportation, is handled in my department, under my supervision." Then he will repeat that familiar but indefinite statement: "Our figures show what it really costs us to operate."

There are thousands of the Smith type of executive, men who think they are keeping costs, although they may have an incomplete system of accounts. The other type is as yet a comparative rarity but his kind is bound to increase. One of the by-products of the work of the Operation and Maintenance Committee may well be the transformation of the Smiths into Johnsons, with benefit to themselves as well as to their employers.

After all, this study we are undertaking is in effect an exploration of the science and art of management, as applied to motor transportation. To the members of the Committee, records or accounts are tools, and tools only. Our purpose is to find out how to use these tools to best advantage.

The first problem of the Committee has been to harmonize what I may call the "service" and the "commodity" points of view. These cannot be separated entirely, yet they seem to be the cause of criticisms of our program that deserve to be met fairly and squarely.

The service point of view is that of the commercial motor-transport organization. It would be held by common carriers and contract carriers of passengers and freight. These carriers or organizations are likely to be influenced by the accounting and other practices developed by the steam and electric-rail systems. In many cases they are required to copy the rail-system methods when they come under regulatory authorities. In others the rail influence is powerful, because of ownership or of the previous training of managers and engineers.

If the motor-vehicles are operated by a company whose general business is the manufacture or marketing of a tangible product, the commodity point of view appears. It is held generally by what may be called the departmentalized form of motor-transport organization. Naturally this is governed to a major extent by the type of business of which it is a department.

The separation between the service and the commodity points of view and between the types of operation which may be described by the two terms is clear enough for the discussion which follows, even though it is easy to find examples of mixture. A taxicab company, although a commercial operator and seemingly in the service group, may have little in common with the local trolley-company. On the other hand, a public utility such as an electric light and power company is a hybrid, since its motor-vehicles are of the departmentalized variety, although the company itself may be considered in the service class, and certainly "enjoys" the regulatory status of its neighboring trolley property.

Now, to indicate the bearing of the two viewpoints or kinds of operation on the problem before the Committee. As regards financial records, the emphasis of the commodity group has been on cost keeping, what the individual vehicle does, or essentially the breaking down of the total of individual items of expense, such as tires and gasoline, into component parts designed to serve as units of performance comparison. If any general accounts have been kept they are usually incomplete, with little or no attention paid to items of administrative or overhead expense for which the motor-transport department might in all fairness be charged.

#### Basis of Emphasis on Cost Keeping

Just what is the cause of the emphasis on cost keeping? It probably is due to a combination of circumstances no one of which is more important than the others. There was the piecemeal nature of the first installations, with one vehicle purchased at a time and each unit considered the subject for a self-contained system of financial and performance records. Then this method became a habit even after it apparently was outgrown, or at least proved inadequate, when a few pieces of equipment gradually developed into an essential department of the organization. Another and perhaps more important cause of the tendency toward cost keeping is the example set by other departments of the business. It has long been the custom in manufacturing and merchandising business to record detailed information of the expense for a given article or job or process, and thus analyze the expense for such broad items as labor and material into their component parts. The process was fairly accurate so long as the greater part of the expense consisted of labor and material which could easily be allocated to the particular job or product. The distribution or allocation of the remainder could then be subjected to the "elastic technique" characteristic of cost keeping; in other

words, it was a matter of more or less refined guesswork but with the possible inaccuracy kept within reasonable limits.

The shoe is on the other foot, however, in the case of the average service type of operation. Direct labor and expense is a small part of the total expense, even with transportation equipment applied to the commodity type of installation. Hence, the emphasis in railroad accounting and to a considerable extent in that used for commercial motor-transport companies has been on the working-out of the general accounts. Totals were wanted rather than components, and the income statement, it was thought, showed whether the expense was too high. If that happens there are conventional methods of pruning that can easily be put into effect.

As a matter of fact, with lower profits and strict regulatory control, the transportation companies are making more and more effort to detect and localize losses and inefficiencies, to compare different methods of doing the same work and to balance revenue against expense for a given route or job, by the use of cost-keeping methods taken over from outside businesses. So far this work has been undertaken only by a few pioneers, however.

With the emphasis of commodity accounting on cost keeping and of service accounting on the general or financial accounts, and with each one leaning a bit toward the other, the leaning being somewhat more pronounced on the part of the commodity toward the general accounts, what was our Committee to do?

## Breaking Down Expense

One of the first suggestions was that the items of expense for motor transportation are simple, well understood and generally followed, so that all we need to do is to define and recommend the methods of unit comparison, such as the ton-mile, to be followed. A little investigation showed a wide variation in the items different people include under the head of operating expense. The same items or accounts had many and sundry meanings, according to their user. Moreover, it was pointed out that the breakdown of expense for comparison would be of little value unless there were a definite starting point.

This situation suggests a standard list of items, the total of which shall be known as operating expense. In other words, a list to correspond with the classification of operating expenses promulgated by the Interstate Commerce Commission for steam railroads. Our efforts would be confined to the operating-expense classification because that is of interest to every user of business motor-vehicles. It is the most important also, as well as the most pressing.

The list or classification of this kind has been well defined as

> A comprehensive plan designed to reveal the economic effect of every factor essential to a true statement of the progress or condition of an enterprise.

Its main advantage is that, if followed, intelligent comparisons can be made of detailed results secured by different companies and at the same time each company can break down the individual accounts for any form of cost keeping or analysis desired so long as the integrity of the primary accounts is not impaired.

## The Ideal Classification

The ideal classification would be one in which each general and primary account could be used for cost-keeping purposes merely by an arithmetical process such as division to obtain fuel expense per mile operated. The possibility of or the necessity for thus breaking down the accounts would be considered consciously and unconsciously in the formulation of the classification. It might even be desirable to suggest subaccounts that would have the effect of "tying-in" a cost-keeping system with the classification. Thus the matter of cost keeping would be given the most careful considertaion with a view to assuring the use of the classification for what some believe would be its most valuable purpose.

With the idea of such a classification advanced, the next step is to determine whether it can actually be worked out in a form to make it of practical value to the users of motor-vehicles. Obviously it should represent the best practice, the best thought, of those engaged in operation or conversant with its requirements. We must first find out what the best practice and thought is. The next move might be the drafting of a tentative classification for general study and criticism. But the draft should be undertaken, in my opinion, only if it seems justified by the information collected.

We are now collecting information primarily from the large-scale installations. We are, of course, covering all the different kinds of commercial motor-transportation organization and also the departmentalized installations in which the motor-vehicle is an important feature, because the expense of delivery is a substantial portion of the gross earnings or on account of the application of portable power-units.

## Broad Cooperation and Elaborate Basis Needed

The associations of truck owners, coal merchants, laundrymen, oil and fuel companies, and the like have been notified of what we are doing, asked for the results of any accounting work they may have done or may have in view and invited to cooperate with our Committee. In this way and by direct contact with a large number of installations we hope to learn the peculiar requirements of all the leading users insofar as they may affect accounting procedure. The response to our requests for information has been most generous and cordial.

A mere list of account titles does not meet the needs. What we need most is to know what is behind the accounts, how they are used, what they mean; in short, definitions, explanations, instructions, philosophy.

After the large-scale classification has been formulated, it will be easy to recommend one or more condensed versions, so arrangd that the main elements of the classifications will be alike and comparable, the simpler requirements of the smaller operators being recognized at the same time. This method of building a foundation on an elaborate scale has been followed in formulating the railroad classifications and is believed by those who have had long experience in classification committee work to be the only practicable procedure. Accountants seem to have no clear rule for defining or testing the constitution of an account. What importance the charges represented must have to justify being placed in one account is largely a matter of judgment or perhaps of accounting custom and precedent. Each account is a signal, it is agreed, for use in rooting out past mistakes and in making plans for a brighter future. The charges included in any one account should be limited, practice seems to indicate, to business facts or transactions of the same kind, so that every value reflecting the result of an action by an executive should be recorded in an account. This means, apparently, that there should be as many accounts as there are "actions" by the executive, and is of practical use only when the meaning of an action is clearly defined. A more definite guide is the suggestion made by one authority that no account should include more than 5 per cent of the total operating expenses. Certainly an account should not be a general blur devoid of all significance because essential distinctions are merged. At the same time refinements should be avoided. Separate accounts should not be recorded of information needed only occasionally. It would be better when the data are required to take them from the subsidiary records, thus eliminating a condition whereby savings secured by the refinements are less than the accounting cost involved.

## Definitions and Measurement Units Important

The names or titles of the primary accounts will need special attention. Some of those commonly used are either ambiguous or deceptive, not signifying the kinds of transaction or valuation supposed to be included. The definition of

the various accounts is, of course, the principal task before the Committee. This is just as essential as their arrangement in a standard order to be followed in all reports and statements.

Cost keeping, viewed as the breakdown of the primary accounts, seems at first sight extremely complicated. But this may be for want of principles to serve as a foundation. There is much confusion also because of the attempt to measure performance by means of the accounts when it should be kept entirely separate. Whether a vehicle is up to standard as regards fuel and lubricant consumption can be settled by keeping records of mileage and amounts, but these records need not be tied in with the dollar values which are the essential feature of accounting records.

The prime difficulty in breaking down maintenance accounts for cost-keeping purposes, and these are the accounts in which such breakdown is most needed, is the lack of a simple and commonly recognized unit of measurement for repair work actually performed. It seems fairly easy to total the dollars spent for repairs, although even this may be open to question. Miles run may tell roughly what has been taken out of a vehicle or a group of vehicles, but how much does this number of miles indicate as to the repair work put back into them? Value of material applied, an even rougher index, is of doubtful merit because it seesaws greatly with the expense for repair labor.

Our investigation has revealed at least six methods in the nature of cost keeping by which the labor and materials expense usually included in the repairs account can be distributed. I have listed these below, not in the order of their importance but merely to show what the Committee has to face:

(1) Cause of work
  (*a*) Normal wear-and-tear
  (*b*) Collisions
(2) Vehicles
  (*a*) Individual units
  (*b*) By groups, as make, size, job or route
(3) Kind of Work
  (*a*) Preparation, as greasing or cleaning
  (*b*) Routine, as inspection or adjustment
  (*c*) Emergency, as failures or accidents
  (*d*) Overhaul, complete rebuilding
(4) Where Work Is Done
  (*a*) Own shop or garage
  (*b*) By other operators
  (*c*) Outside shops or service stations
(5) Parts or Units Repaired
  (*a*) Chassis
  (*b*) Engine
  (*c*) Body and other divisions as desired
(6) Fundamental Distribution
  (*a*) Materials
  (*b*) Labor
  (*c*) Miscellaneous

This is only a partial list but it illustrates the complexities of the problem. At least one company uses method (2) for shop records, the items being arranged in the accounting department according to method (5).

Our task, as I see it, is to study the best practice in accounting and cost keeping followed by users of motor-vehicles, with the purpose of analyzing it and making the principles derived or other results freely available for the benefit of the public, the users of business motor-vehicles and finally the manufacturing industry which is vitally concerned in the efficient utilization of its product.

## WORLD MOTOR-TRANSPORT CONGRESS

### Society's European Trip Will Include Sessions Held in London in November

In addition to the conferences that will be held in London in connection with the Commercial Motor-Transport Exhibition, members of the Society will attend the sessions of the World Motor-Transport Congress shceduled for the dates immediately preceding the Commercial Motor-Transport Exhibition.

The tentative plans for the European Trip indicate that the party will sail on the Berengaria on Nov. 8, arriving in time for the opening of the World Motor-Transport Congress, leaving England on the Aquitania on Nov. 26 and arriving in New York on Dec. 3. Many members will of course leave on earlier or return on later boats, but one of the reasons for making the trip is the personal contacts established. Society members who participated in the trip to England in 1911 made friendships that have been the most enjoyable part of their professional life.

Special invitations to join the party have been sent to members of the Society interested particularly in automotive transportation. Other members who would like to consider the desirability of making the trip should communicate with the Society's office in New York City to receive more complete information.

Among members who are seriously considering attending the Commercial Motor-Transport Exhibition and the World Motor-Transport Congress are Vice-President W. G. Wall, Councilors J. F. Winchester and Ernest Wooler, and B. B. Bachman, David L. Bacon, N. D. Ballantine, F. J. Campbell, John R. Cautley, A. W. S. Herrington, F. C. Horner, Neil MacCoull, R. E. Plimpton, and J. Webb Saffold.

Plans decided upon for the World Motor-Transport Congress were outlined recently as follows in *Commercial Motor* (London):

> It is claimed for the World Motor-Transport Congress, which is to be held in London in November, that it will be an event of first-class importance. Congresses under this title have been held on three occasions in America, but the forthcoming gathering may be fairly regarded as the first of a new and more generally valuable series of events which, in course of time, may exert very powerful influence in favor of the further development of road transport throughout the world.
>
> The Congress to be held in London this autumn will consist of six meetings, which will take place on the mornings and afternoons of Nov. 14, 15 and 16. On Nov. 17 the delegates at the Congress will pay an official visit to the Commercial Motor-Transport Exhibition at Olympia.
>
> The Society of Motor Manufacturers & Traders, Ltd., is acting as organizer of the Congress under the auspices of the Bureau Permanent International des Constructeurs d'Automobiles, which is a body formed to regulate certain international matters affecting the motor industries of all manufacturing countries.
>
> Although the Congress will take place in London, it will be of an entirely international character. Equal opportunity will be given to all nations to be fully represented, to submit written contributions and to take part in the discussions on the selected subjects. It is anticipated that further Congresses will subsequently be held in the capitals of other countries, the programs for these later events depending on the decisions which will be arrived at in November as to what subjects require further investigation and what information or statistics ought to be collected as a basis for later discussion.
>
> In the present instance, six main headings for discussion have been agreed upon. These are as follows:
>
> (1) Road construction and improvement in relation to the development, efficiency and economy of road transport
> (2) Mechanical road-transport as an instrument of development of world resources
> (3) The necessity for cooperation between road and rail transport
> (4) The development of motor-vehicles suitable for service on bad roads and for cross-country use

(5) The improvement of facilities for international travel by road
(6) Fuels and fuel supplies for road motor-vehicles

Invitations are being addressed to the Governments of all nations and to important organizations concerned with inland transport throughout the world, and all who nominate delegates are invited to send in written contributions under any of the headings mentioned above. These contributions will be distributed to all delegates well in advance of the meetings. Consequently, it will not be necessary to take up time at the actual meetings by the reading of papers. Those who have submitted papers will be asked to summarize the main points they wish to bring forward and so open a general discussion, in which it is hoped that not only those who have information to give but those who desire to obtain information will take equal part.

It has been suggested that the forthcoming Congress may be fairly regarded as the first of a new series of gatherings of this kind. Previous Congresses under the same name have been attended largely by people directly connected with the business of motor manufacture and distribution. Such interests will, of course, be again represented in London, but a very large proportion of the representation will be official. Arrangements have already been made under which all parts of the British Empire will be officially represented.

## BIBLIOGRAPHY FOR VEHICLE OPERATORS

### Research-Department List of Reference Articles Published by the Society

Since the organization of the Operation and Maintenance Committee of the Society there has been a pronounced development of interest among the members and many non-members who are engaged in motor-vehicle fleet-operation and maintenance. Papers and articles relating to this branch of the automotive industry and business in general throughout the Country have been read at meetings of the Society and published in THE JOURNAL. To give the members direct references to these, the following compilation has been made of articles published by the Society.

Brennan, E. J.—Motorized Railroad Equipment—THE JOURNAL, June, 1924, p. 656; and September, 1924, p. 195. Reprinted in TRANSACTIONS, Vol. 19, Part I, p. 582.
Deals with general subject of costs.

Cain, James W.—Motor Rail-Cars—THE JOURNAL, October, 1924, p. 317. Reprinted in TRANSACTIONS, Vol. 19, Part II, p. 438.
Gives data on operating costs.

Fairbanks, William H.—Operating Costs and Economic Life of Motor Vehicles—THE JOURNAL, March, 1926, p. 285.

Favary, Ethelbert—Six-Wheel Truck Construction and Operation—THE JOURNAL, April, 1925, p. 427. Reprinted in TRANSACTIONS, Vol. 20, Part I, p. 197.
Gives data on the cost of transportation.

Guernsey, C. O.—Requirements in Rail-Car Design—THE JOURNAL, July, 1925, p. 74; and December, 1925, p. 609. Reprinted in TRANSACTIONS, Vol. 20, Part II, p. 525.

Hatosy, F. E.—Inspection, a Means for Economical Motor-Truck Fleet-Maintenance—THE JOURNAL, June, 1926, p. 637.

Hutt, A. E.—Motorbus Transportation—THE JOURNAL, June, 1924, p. 591; and September, 1924, p. 251.
Gives data on maintenance costs.

LaSchum, E. E.—Engineering Brains in Fleet Operation—THE JOURNAL, October, 1923, p. 276. Reprinted in TRANSACTIONS, Vol. 18, Part II, p. 743.
Discusses fleet operating-costs.

Plimpton, R. E.—An Analysis of Costs for 10 Years of Fleet Operation—THE JOURNAL, May, 1924, p. 539; September, 1924, p. 206. Reprinted in TRANSACTIONS, Vol. 19, Part I, p. 665.

Power, Eugene—The Cost of Operation and Economic Life of Motor Trucks—THE JOURNAL, July, 1926, p. 59.
A general discussion with charts showing costs per mile for 387 automobiles from 1919 to 1923 and for 2 contrasting types of truck.

Ruprecht, Louis—Cost of Work with Gasoline Motor-Trucks—S.A.E. BULLETIN, May, 1912, p. 44. Reprinted in TRANSACTIONS, Vol. 7, Part II, p. 86.
Deals with truck operating-costs.

Schipper, J. Edward—Motor-Transport-Field Opportunities for the Automotive Engineer—THE JOURNAL, May, 1926, p. 463.

Scott, Joseph—Analysis of Operating Costs of Automotive Equipment—THE JOURNAL, April, 1926, p. 367.
Gives charts and graphs of operating and maintenance costs and economic life of vehicles.

Winchester, J. F.—Observations of a Superintendent of Motor-Truck Fleet-Operation—THE JOURNAL, May, 1924, p. 496; December, 1924, p. 567; and May, 1925, p. 534. Reprinted in TRANSACTIONS, Vol. 19, Part I, p. 597.

## STATE MOTOR-VEHICLE REGULATIONS

### Subcommittee of Operation and Maintenance Committee To Study Variations

After the Operation and Maintenance Committee of the Society was organized this year, one of the principal studies it planned to undertake is a comparison of the regulations in force in the several States, and eventually to prepare recommendations that would make such regulations more uniform with the object of relieving motorcoach and motor-truck manufacturers of the necessity for changing constructional features of the vehicles to meet the varying requirements in different States.

Another phase of the study will be to submit recommendations regarding existing regulations that make it difficult for operators to maintain vehicle operation in conformity with those regulations. An example of this is the requirement that the license issued for a given vehicle must bear the engine number of that vehicle. An operator who replaces an engine in a vehicle must either take out a new license for the new engine number or change the number on the replacing engine to correspond with the existing license. It is felt that State regulations can be modified to permit an operator to make a complete engine-change without requiring a new license or delaying unduly the operation of the vehicle.

This Subcommittee held its first meeting in New York City on June 29, at which time it was agreed that the large representation of the motor-vehicle operating-engineers in the Society justifies the Subcommittee's taking up this work regarding State and National regulations when they affect motor-vehicle operation and maintenance. It was also felt that the subject of taxes might well be included in the scope of the Subcommittee's work. Among the items discussed, the following were agreed upon as of immediate interest:

The matter of present State requirements for registration of engine numbers which make it difficult to interchange engines between vehicles, a procedure which is often required in fleet maintenance.

Variation in requirements for the location and color of motorcoach marker-lights, which causes confusion in the operating of interstate lines and in the shifting of vehicles from one State to another.

(*Concluded on p.* 192)

# Standardization Activities

## OIL AND GREASE-CUP REVISION

### Subdivision To Revise Present Specifications by Adding Shank Dimensions

A Subdivision of the Parts and Fittings Division was recently organized to consider the revision and amplification of the present S.A.E. Recommended Practice for Oil and Grease-Cup Threads. The present specification, printed on p. C57 of the March, 1927, issue of the S.A.E. HANDBOOK, includes only the nominal thread-size. It has been suggested that consideration be given to adding dimensions for the threaded-shank length and the width across flats of the hexagon shank.

While no meeting of the Subdivision has as yet been held, the chairman, G. L. McCain, of Dodge Bros., Inc., has prepared the following revision of the table as a basis for discussion:

| Nominal Thread-Size, In. | Threads per Inch | Width Across Flats of Hexagonal Shank, In. | Threaded-Shank Length, In. |
|---|---|---|---|
| 1/8[a] | 27 | 7/16 | 3/8 |
| 1/4[a] | 18 | 9/16 | 7/16 |
| 3/8[a] | 18 | 3/4 | 1/2 |
| 1/2[a] | 14 | 7/8 | 9/16 |
| No. 10 | 32 | 7/16 or 3/8 | 1/8 |
| 1/4 | 32 | 7/16 or 3/8 | 3/16 |
| 5/16 | 32 | 7/16 or 3/8 | 1/4 |

[a] Pipe thread.

This suggested specification is now being circularized to the principal manufacturers of oil and grease-cups and to the Subdivision members with the request that they submit their comments and suggestions so that the first Subdivision meeting may have some definite figures to discuss.

The Subdivision has the following personnel: G. L. McCain, Dodge Bros., Inc., chairman; A. K. Brumbaugh, White Motor Co.; R. J. Gits, Gits Bros. Mfg. Co.; A. W. Reader, Bassick Mfg. Co.; and W. J. Outcalt, General Motors Corporation.

## NEW S.A.E. STEEL RECOMMENDED

### Subdivision Submits High-Manganese, High-Sulphur Steel Specification

Discussion at the last meeting of the Iron and Steel Division held in Metroit on March 29 indicated the need for high-manganese, high-sulphur steel specifications in view of the large usage that this type of steel is finding in the industry.

A Subdivision consisting of R. B. Schenck, Buick Motor Co., chairman; H. W. Graham, Jones & Laughlin Steel Co.; W. G. Hildorf, Reo Motor Car Co.; W. R. Shimer, Bethlehem Steel Co.; and E. W. Upham, Chrysler Corporation, after giving the matter thorough consideration, recommends the following specification for approval by the Iron and Steel Division:

PROPOSED NEW STEEL

| Element | Per Cent |
|---|---|
| Carbon | 0.10-0.20 |
| Manganese | 1.25-1.55 |
| Phosphorus, maximum | 0.05 |
| Sulphur | 0.08-0.13 |

This proposed specification is being circulated among steel mills and manufacturers of automotive vehicles for comment and criticism, and a summary of the replies will be presented to the Subdivision for consideration at its next meeting.

The following excerpt from the minutes of the Subdivision meeting, held in Detroit on June 22, gives the reasons which led to the adoption of values of the various elements of this steel:

> It was felt that the principal demand at the present time is for a carburizing steel with a carbon range of 0.10 to 0.20 per cent. Steels of this type with high carbon-content are used to some extent, but the demand at present is rather limited. It was agreed that a range of 0.15 to 0.25-per cent carbon is undesirable for most carburizing work.
>
> It was felt that one manganese range instead of two or more is highly desirable in the interest of standardization. Little information seems to be available as to the relative machinability of the two types; namely, 1.00 to 1.30 and 1.25 to 1.75 per cent of manganese. Steels with manganese under 1.20 per cent approach rather closely S.A.E. Steel No. 1120, while steels with over approximately 1.60 per cent of manganese have been reported in several instances as deficient in machinability. It was agreed that a range of 1.25 to 1.55 per cent seems to give the most satisfactory results.
>
> It was decided to specify a maximum phosphorus-content of 0.050 rather than 0.045 per cent as the higher figure should prove perfectly satisfactory and would facilitate the manufacture of this steel by the acid open-hearth process.
>
> A sulphur content of 0.09 to 0.13 per cent was at first proposed, but it was felt that the steel manufacturers should have a range of 0.05 instead of 0.04 per cent. Considerable discussion ensued as to whether the range should be widened upward or downward. The 0.14 per cent of sulphur would tend to produce a greater number of manganese-sulphide inclusions; on the other hand, 0.08 per cent of sulphur would tend toward impaired machinability. A range of 0.08 to 0.13 per cent of sulphur was finally agreed upon. It was the opinion of the Subdivision that a 0.10-per cent minimum sulphur-content would necessitate too high a maximum value for this element.

## REVISED BATTERY SPECIFICATIONS

### Demand for Thicker Plates Necessitates Larger Motorcoach-Battery Dimensions

The formulation of the present motorcoach-battery specifications on p. B23c of the March, 1927, issue of the S.A.E. HANDBOOK was based on the ratings and dimensions of the 5/32-in. plate batteries made by various companies. Since that time the development of the ¼-in. battery-plate and the increase in demand for the thick-plate type has necessitated a revision of the specifications.

The dimensions of the thicker-plate battery are of necessity somewhat larger, although in general the present dimensional specifications cover this type of battery.

The Subdivision on Motorcoach Storage-Batteries, at a meeting held in the Old Colony Club, Cleveland, on July 18, found it advisable to recommend to the Electrical Equipment Division a revision of the maximum over-all length of batteries Nos. 24 and 26 and an increase in the maximum height of all batteries, to permit the use of stronger cases.

Owing to the increase in battery efficiency it was found possible to recommend raising the minimum battery-ratings for most of the sizes. A new specification to cover large-size batteries was added under the number 27 to provide mounting dimensions for motorcoach builders using this type of battery.

The following table shows the revised specifications to be presented to the next meeting of the Electrical Equipment Division for approval:

PROPOSED REVISED STORAGE-BATTERY SPECIFICATIONS

| Battery No. | 21 | 22 | 23 | 24 | 25 | 26 | 27 |
|---|---|---|---|---|---|---|---|
| Number of Cells | 3[a] | 3[a] | 3[a] | 6[a] | 6[b] | 6[a] | 6[a] |
| Minimum Capacity at 8-Hr. Rate, amp-hr. | 110 | 129 | 178 | 88 | 88 | 104 | 118 |
| Minimum Current for 20 Min., amp. | 135 | 157 | 225 | 112 | 112 | 140 | 160 |
| Maximum Over-all Dimensions, in. | | | | | | | |
| Length[1] | 17 | 19⅛ | 25⅞ | 26⅞ | 20⅞ | 32½ | 34 |
| Width | 7⅝ | 7⅝ | 7⅝ | 7⅝ | 9⅝ | 7⅝ | 7⅝ |
| Height | 10⅞ | 10⅞ | 10⅞ | 10⅞ | 10⅞ | 10⅞ | 10⅞ |

[a] Side-to-side assembly of cells.
[b] Double-row, end-to-end assembly of cells.
[1] The over-all end-to-end length includes handles, not hold-down devices. The handles and the hold-down devices shall be attached only to the ends of the case. Terminals and connections shall not extend above the handles; the latter shall be the highest point.

*Note.*—It is recommended that where the method of installation makes the handling of heavy 12-volt single-unit batteries difficult, two 6-volt units of similar capacity be used.

The note with reference to the use of two 6-volt units in place of one 12-volt was approved after considerable discussion, it being felt that no definite size could be specified as the maximum for a single unit.

The Subdivision personnel is as follows: A. K. Brumbaugh, White Motor Co., chairman; C. T. Klug, Willard Storage Battery Co.; G. L. Kyle, USL Battery Corporation; L. E. Lighton, Electric Storage Battery Co.; S. W. Mills, Pierce-Arrow Motor Car Co.; J. L. Rupp, Westinghouse Union Battery Co.; and T. E. Wagar, Studebaker Corporation of America. All members were present except Mr. Mills, who was abroad.

## SURVEY OF FLYWHEEL-HOUSING LIMITS

### Opinion of Clutch and Engine Manufacturers Determined on Present Standard

Some time ago a suggestion was received from one of the leading clutch manufacturers to the effect that the Transmission and Engine Divisions consider setting closer limits between the engine flywheel-housing and the transmission clutch-housing in unit powerplant mountings. Transmission engineers were of the opinion that tolerances for the eccentricity could be reduced as well as tolerances for the bore to eliminate some of the undesirable looseness which is permitted by the present limits. It was also suggested that the holes in clutch housings be specified as 3/64 in. larger than the cap-screw instead of 1/32 in. larger as now specified.

Before taking any definite steps it was considered advisable to canvass all manufacturers of clutches and engines including passenger-car and motor-truck manufacturers building their own powerplants to determine to what extent the present S.A.E. Standards are used and what difficulty, if any, had been experienced due to warpage, springing under load or similar conditions.

To determine the attitude toward these changes and the manufacturers' opinions as to the value of the suggestions, a letter was forwarded to manufacturers mentioned above with a return-form containing the following questions for which answers were requested:

(1) Do you use the present S.A.E. Standard dimension for flywheel housings?
(2) Are the present limits satisfactory?
(3) Do the present limits permit misalignment?
(4) Have you experienced difficulty due to warpage, springing under load or similar conditions?
(5) Do you "true" flywheel housings for assembly?
(6) Do you approve making holes in the clutch-housing 3/64 in. larger than the cap-screw instead of 1/32 in. larger as now specified?

The accompanying table records the opinions as received and this summary together with all comments received will be presented to the Transmission and Engine Divisions at the next meetings for their decision as to whether any further action should be taken looking toward revision in accordance with the suggestion received. From the comments submitted on the return-form it appears to be the consensus of opinion that if the present limits are adhered to no difficulty will be encountered and that the standard dimensions as given are in themselves satisfactory.

## REPORT ON MATHEMATICAL SYMBOLS

In the report of the Mathematical Symbols Subcommittee of the Sectional Committee on Scientific and Engineering Symbols and Abbreviations, which was printed in the July issue of THE JOURNAL on p. 4, several typographical errors occurred. These are listed below.

In Section 1.2, = in the second line should have been —.

In Section 3.3, $\alpha$ should have been used in giving the value for $l$ instead of $\infty$. This error was, of course, obvious.

In Section 5.3, the expression signifying summation should have been written as follows:

$$\sum_{i=a}^{b}$$

In Section 5.6, the solidus in the second member of the equation in the last line should have been omitted.

## RUBBER PARTS TO ABSORB VIBRATION

### Society Cooperating in Study of Tests for Mechanical Rubber Parts

On several occasions inquiries have been made of the Society regarding any tests that may have been adopted for determining the life and other qualities of mechanical rubber parts such as engine support-blocks, universal-joints, spring-shackle blocks, and door shock-pads. Inasmuch as no such tests have been developed, the subject was discussed with the American Society for Testing Materials with which the Society cooperates in connection with material-testing specifications. Accordingly, the Subcommittee on Rubber Products That Absorb Vibration is being organized under Committee D-11 for Rubber Products of That Society and it is anticipated that the Subcommittee will develop many new technical data that will be of great value to the automotive and other rubber-using industries. As the work of this Subcommittee progresses it will be reviewed in subsequent issues of THE JOURNAL.

| Manufacturers of | Number of Companies | Questions 1 Yes | 1 No | 2 Yes | 2 No | 3 Yes | 3 No | 4 Yes | 4 No | 5 Yes | 5 No | 6 Yes | 6 No |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Clutches | 9 | 4 | 3 | 6 | 1 | 6 | 2 | 2 | 3 | 5 | 2 | 5 | 4 |
| Passenger-Cars Having Own Engines | 23 | 10 | 11 | 10 | 3 | 6 | 4 | 13 | 6 | 5 | 7 | 6 | 8 |
| Motor-Trucks Having Own Engines | 7 | 5 | 2 | 4 | | 1 | 2 | 2 | 2 | 2 | 1 | 3 | 2 |
| Engines | 7 | 5 | | 7 | | | 4 | 6 | 1 | 4 | 2 | 5 | 1 |
| Total | 36 | 24 | 16 | 27 | 4 | 13 | 12 | 23 | 12 | 16 | 12 | 19 | 15 |

# Production Engineering

## ELECTRIC ARC-WELDING PAPERS

### Requirements Governing Awards in Competition on New Developments and Applications

As stated in THE JOURNAL for May, the American Society of Mechanical Engineers has accepted the custody of $17,500 to be awarded in a world-wide competition for the best three papers disclosing advance in the art of arc welding. It is believed that this art presents opportunity for great development. In administering this trust, the American Society of Mechanical Engineers will appoint a committee of judges.

Any one, in any country of the world, may compete for the prizes. All papers must be written in duplicate, in the English language, and submitted to the Council of the American Society of Mechanical Engineers, addressed to the secretary, Calvin W. Rice, 29 West 39th Street, New York City. All exhibits must be in duplicate. Only papers bearing evidence of having been sent before Jan. 1, 1928, will be considered. Any paper arriving after Jan. 31, 1928, will be disqualified.

Every paper should include all necessary drawings, photographs, or other exhibits, so that the disclosure will completely present the suggestions of the competitor. The utility of the suggestions must be shown not only for the application but also for possible use in connection with other designs and for other purposes. The possible advantages and economies of the methods should be made clear, as these will be the chief base upon which winning papers will be selected.

#### PRACTICAL USES MUST BE OUTLINED

The practicability of the suggestions must be evident from the paper. It is not necessary, however, that their actual application be fully shown, if possible uses are made clear. Originality of design is preferable, either in the method of applying the weld or in the design of the welding parts or their arrangement. Designs which are of no practical use will be considered only in case they include suggestions which could obviously be applied in other ways than those suggested. Methods of applying the arc-welding art which will improve existing machines, or make commercially possible machines which, in the light of previous engineering, have been regarded as impractical, are specially desired. It is not necessary that all parts of the structure be made of welded metal; it is only necessary that one or more parts be so made.

In case two or more identical suggestions are received, the paper bearing the lowest serial number on receipt by the secretary of the American Society of Mechanical Engineers will have precedence. Because of this rule, competitors are urged to submit their papers as soon as they are prepared.

The Council of the American Society of Mechanical Engineers may withhold any or all awards.

Each paper must be typewritten on one side only of 8½ x 11-in. paper and bound at the top with a protecting cover or jacket. The name and address of the competitor must appear on the front cover. Both copies of each paper must be enclosed within a sealed envelope, on the outside of which are placed the name and address of the competitor and the following legend: "Submitted to the American Society of Mechanical Engineers, 29 West 39th Street, New York City, in competition for the Lincoln Arc-Welding Prize." This sealed envelope must be enclosed within a mailing wrapper. The receipt of each paper will be acknowledged.

The committee of judges will not pass upon the acceptability of any apparatus suggested as a proper subject for this competition.

In submitting papers, each competitor gives to the American Society of Mechanical Engineers the right to publish his paper, should it desire to do so, in any of its several publications. Papers will not be returned to competitors.

The decision of the committee of judges in interpreting the foregoing rules will be final.

It is recommended that in preparing papers the small amount of literature available be studied to get the fundamental principles which govern re-design.

It is the purpose of the American Society of Mechanical Engineers to present the awards in 1928 at its Spring Meeting. All competitors will be properly notified of the action of the judges.

#### AMOUNTS OF AWARDS

The awards will be made to the authors of the most valuable papers suggesting means and methods for carrying out the purposes of the competition and will be in the amounts of first, $10,000; second, $5,000; and third, $2,500.

It is suggested that members who wish to enter this competition communicate with the Secretary of the American Society of Mechanical Engineers at 29 West 39th Street, New York City.

#### BIBLIOGRAPHY

Although comparatively little has been published on the art or practices of arc welding, the following references are furnished by the Research Department of the Society of Automotive Engineers:


Automatic Arc-Welding Finds Many Uses in Automotive Production—W. M. B. Brady and W. L. Warner—*Automotive Industries*, June 24, 1926, p. 1092.

Body-Welding Methods Cut Labor Costs to Minimum—J. Edward Schipper—*Automotive Industries*, April 9, 1925, p 652.

Electric Arc-Welding in Airplane Construction—A. G. Bissell—*Aviation*, March 29, 1926, p. 446.

Many Automotive Production Problems Solved by Welding Processes—P. M. Heldt—*Automotive Industries*, Sept. 24, 1925, p. 506.

Electric-Welding Processes in the Automobile Industry—P. Fassler—*Automotive Industries*, Aug. 26, 1926, p. 343.

Metal-Welding Process Important Step in Auto Production—*The Automotive Manufacturer*, June, 1925, p. 10.

Observations on Welding All-Steel Automobile Bodies—Joseph W. Meadowcroft—*Motor Vehicle Monthly*, December, 1925, p. 27.


The Research Department also indicates the relative amount of welding used on open and sedan bodies by the electric-spot, the acetylene-gas and the arc-welding methods.

RELATIVE AMOUNT OF WELDING ON AUTOMOBILE BODIES

| Type of Body | Open | Sedan |
|---|---|---|
| Acetylene-Gas, in. | 70 to 80 | 170 |
| Electric-Arc, in. | 10 to 12 | 11 to 12 |
| Electric-Spot, in. | 1,000 | 1,600 |

## THE PROBLEMS OF COST CONTROL

### Materials, Equipment and Labor Comprise Production Engineer's Study Elements

The province of the production engineer is primarily composed of three elements. Materials come under his control carefully specified and designated for a specific purpose. What he is to do with them is preordained in a ceremony in which he may or may not have had a part. Labor comes in variable man-units of alleged ability and constancy applied more or less energetically for stated periods of time.

Equipment, however, is more stable, more fixed in its habits and amenable to control, sometimes unfortunately too persistently permanent.

These elements entail certain items of expense which comprise manufacturing cost and by which the ability of the production engineer is judged. An intelligent interpretation of cost figures is, therefore, of very considerable interest to the production engineer in order that he may recognize and understand completely the shop conditions which are represented in such a statement. There are many indicative ratios in terms of quantities produced, number of employes, man-hours, floor space, and the like, which take on a new significance when expressed in the dollars and cents of manufacturing costs.

The idea of volume production at any cost is no longer permitted, if it ever was. Slower expansion has brought about the necessity for reducing costs and for earning more profit per unit of production. The quality of materials and workmanship, however, must not decline. It must improve. It is, therefore, clearly up to the efficiency of labor and more especially up to the equipment and methods element to show sufficient economy to offset whatever extra cost is involved in maintaining or improving quality, and then to earn a net gain.

Many of the physical problems in the material element are to the production engineer an undesirable legacy which cannot be avoided and of which he is too often critical. They refer not only to the properties of various materials specified but also to the designs of parts required to manufacture, as well as to the grotesque contortions sometimes necessary in assembling. But well within the production engineer's realm are the matters of handling and inspecting the materials and parts through production.

### Labor Problems Only Partly Solved

The problems of the labor element have been partly solved, but they too extend outside the realm of the production engineer. Wherein they have been under his control, getting the most results per labor dollar, he has done a creditable job. On the whole, however, industrial relations are broader, and perhaps deeper than he cares to control. They deal with personal behavior due to mental reactions from social conditions brought about by economic necessity. The average production man is no hyfalutin' 'ologist. Granting then that the progress to be made in labor efficiency is relatively small and, with a more extensive use of automatic equipment, becoming relatively smaller, he must work out his salvation in terms of equipment and the choice of methods that the various types of equipment signify.

Such physical conditions are well known but the complexion which cost analysis gives them is too often thought to be as false as the drug-store variety on a pretty face. Dependable costs, and they are worse than worthless if they are not dependable, interpret the physical conditions and factors with which the production engineer is dealing in terms of dollars and cents which affect the profit, the ultimate idea in manufacturing goods for sale.

Take then some of the physical ratios and translate them into cost ratios. The unit of production is the common divisor. Direct labor in man-hours has been divided by production quantity in piece-work, production bonus-schemes and in almost every conceivable way to establish a control. Yet, here are some of the varied interpretations of a possible case. The job is one requiring a measure of skill on the part of an operator running a machine, for instance a turret lathe. A 40-cent boy produces 110 pieces in the same time that it takes a 60-cent man to produce 150 pieces. The direct labor-cost favors the boy on the job. Under a production bonus-plan with the quantity of 100 as a base figure, the boy would be even more preferable. A percentage overhead to direct labor still gives him the advantage, but as soon as complete cost analysis shows that the machine is costing $2 per hr. regardless of who runs it, the more competent operator's cost per piece is shown in its true light, and, when factors of inspection and supervision are considered with more attention required for the less competent operator, the greater production by the considerably more expensive man is far ahead. This, then, shows the opportunity for fallacy. It also shows what a thorough understanding of cost analysis may mean.

With dependable production economy, the percentage of direct labor that is operating under the bonus plan to that operating under the day work plan is, of course, significant. The relation of the amount of indirect labor employed to the direct labor indicates a probable result but is more truly indicative when rates and the expense of service equipment used are also considered in conjunction with the actual work performed.

### Floor-Space an Important Factor

The floor-space factor has been one of considerable discussion and the saving of floor space the object of strenuous endeavor. How important it is may be determined by considering the investment in equipment that will produce in 50 sq. ft. what previously required 500 sq. ft. With rent and building expenses costing in the neighborhood of $1.50 to $2.00 per sq. ft. per year, the annual net floor-space saving ranges from $675 to $900. This is relatively a small factor in consideration of the investment required, but under crowded conditions the value of the space released is far in excess of the figures placed on it by cost analysis.

Reduction of small-tool and supply items, such as oil for lubrication, can be made to the benefit of each expense item but often to the detriment of maintenance costs and possibly to the actual loss of production. The attitude of bearing hard on all overhead items in a special drive for economy can be too easily carried not only to a point of diminishing returns but also to the extent of losing money by aggravated conditions in other items. This is particularly true in the case of comparing by percentage the overhead, which includes the fixed charges on equipment, with direct labor. As truly as machinery is taking over the functions of labor and releasing it for use in greater production or to a higher purpose, so truly must the burden expense which includes the cost of carrying and operating such equipment multiply in proportion to the direct labor-costs. It is reasonable to look for evidence of improvement, then, in the increase of some items of expense and to ratios that would have been frightful to the managers of 10 or even 5 years ago.

The production engineer who can intelligently interpret costs to the betterment of his control will find a further use for them in breaking such facts down into understandable units for the benefit of foremen and even for the workers. Here exists a real desire to have a part in improvement, for which such periodical reports form an incentive. Here, too, is a desire to improve the conditions under which the workers operate, and to contribute to a greater ultimate profit in which they may reasonably expect to share. Best of all is impressed the idea that progress is measured in a greater ratio of power consumed and equipment used per worker.

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## HUMAN ELEMENT IN INSPECTION

### R. R. Todd[1] Emphasizes an Important Factor in the Production of Automobiles

No matter how intricate a part may be, it is possible to design gages that will function without much difficulty. But gages do not take into account the most important factor in the use of a gaging system, the human element. For example, gears, axles, engines, transmissions, and other parts are liable to develop noise. It may be practically impossible, to get, say, a differential assembly through inspection one day, yet the next day the gears will be acceptable although

(*Concluded on p.* 199)

[1] M.S.A.E.—Chief inspector, Oakland Motor Car Co., Pontiac, Mich.

# Automotive Research

## INSTRUMENT RESEARCH AND DESIGN

### Progress Reports to Riding-Qualities Subcommittee Cover Theory and Practice

Research may take one of two directions: it may explore the details of principles or processes in an attempt to define more exactly underlying fundamentals, or it may adapt to general practical usage a finding of science that has been confined to the realm of theory. In the three contributions made to Automotive Research this month, both types of research are represented. All of them deal with riding-qualities, a subject so complex, abstruse and of such timely interest as to assure for them a welcome from all students of the subject. They are contributed by investigators whose accomplishments in this field bring them a ready hearing; C. M. Manly[1], Dr. Benjamin Liebowitz[2] and R. W. Brown[3]. This material was presented at a meeting of the Riding-Qualities Subcommittee of the Research Committee held during the Semi-Annual Meeting at French Lick Springs, Ind.

One article, Contact-Accelerometer Gap-Error, exemplifies the more theoretical research. In it Dr. Liebowitz develops a refinement of operating methods for contact accelerometers, taking readings from the lower contact; this being in addition to that set forth by him in a previous article in THE JOURNAL[4]. In his earlier publication Dr. Liebowitz analyzed the effect of gap-clearance and derived a gap-error formula. The present note defines more exactly one of the quantities used in the formula.

The second article reports research directed toward laboratory verification of deductions arrived at in Dr. Liebowitz's theoretical research. C. M. Manly, C. B. Veal[5] and Dr. Liebowitz have been studying for some time the problem of devising an apparatus to test accelerometers. Under the title, Accelerometer Testing Apparatus, Mr. Manly and Dr. Liebowitz here summarize the results of the development work. In brief, the testing apparatus constructed incorporates an attempt to avoid the uncertain errors of any crank-motion by obtaining harmonic motion through the rotation of eccentric weights mounted on a spring-board, thus obtaining the advantage of extreme simplicity, minimum cost and at the same time avoiding the errors incident to lost motion by employing a molecular hinge.

Recent studies made by R. W. Brown, at the laboratories of the Firestone Tire & Rubber Co., may be classified as the second or more practical type of research. Basing his development work on his earlier investigations into fundamental principles of riding-qualities instrumentation, Mr. Brown has designed an accelerometer destined for practical road-usage. A progress report of riding-qualities research along these lines is incorporated in the third item, Riding-Qualities Instrumentation, prepared by Mr. Brown.

## CONTACT-ACCELEROMETER GAP-ERROR

### Study of Formula Leads to More Exact Definition of One Factor Involved

In a previous article[4] an analysis was given of the error due to the inertia of the moving system in the use of the lower or "make" contact of contact accelerometers. In this analysis I derived the formulas:

$$E = (a_m - a_s)/a_m = 2/3 \vee (c/2A) \quad (1)$$

$$E = (a_m - a_s)/a_m = 2/3 \vee [c_1(1 + [k^2/r^2])/2A] \quad (2)$$

The first formula applies to accelerometers whose moving systems move in a straight line, without rotating; the second applies to instruments with *pivoted* moving-systems, which therefore rotate.

In these formulas $a_m$ is the maximum value of the impressed acceleration, and is the acceleration whose measurement is sought; $a_s$ is the acceleration for which the instrument is set; $2A$ is the stroke of the impressed motion, assumed to be simple harmonic; and $c$ is the "gap". In the second formula, $c_1$, is the "effective gap" measured at the center of gravity of the inertia element; $k$ is the radius of gyration of the inertia element about its center of gravity; and $v$ is the distance from the pivot to the center of gravity.

Attention is here directed to the quantity $a_s$, the acceleration for which the instrument is set. It is clear that when the lower contact is used there are two zero-settings of the instrument; ($a$) when the upper contact just "breaks" and ($b$) when the lower contact just "makes." Hence, for any other setting of the instrument, there will be two values of acceleration, according as we take the first or the second of the above-mentioned zero-settings. Let us distinguish between these two values of $a_s$ by calling the first $a_1$ and the second $a_2$; that is, $a_1$ corresponds to the upper-contact zero-setting and $a_2$ to the lower-contact zero-setting. The difference between these two values is sometimes considerable. Now, in deriving the gap-error formula, I defined $a_s$ as identical with the value here called $a_1$, whereas, in actual practice, in view of the difference between $a_1$ and $a_2$, it is more correct to use the value of $a_2$ as the indication of the instrument. The gap-error formula does not apply to the value $a_2$, but only to the value $a_1$. Failure to realize this will lead to erroneous results, as has been learned by experience.

To use the gap-error formula under such conditions, it is necessary to go back to its derivation, wherein the motion of the inertia element was calculated on the assumption that $a_s$ is constant throughout the travel from the upper contact to the lower. For this reason, the formula was restricted to small gaps. Actually, however, $a_s$ varies throughout the motion from $a_1$ to $a_2$, so that a good approximation is obtained if we take $a_s$ as the *average* between $a_1$ and $a_2$.

Therefore, by defining $a_s$ as

$$a_s = \tfrac{1}{2}(a_1 + a_2)$$

instead of as $a_s = a_1$, the formulas are more accurate and may be applied with good results to gaps of fair size.

It should be mentioned that in a few practical applications so far, the value of $a_s = \tfrac{1}{2}(a_1 + a_2)$, when corrected by the formula, gives a result which does not differ very much from $a_2$, even for gaps of fair magnitude. Further analysis is therefore desirable to express the error in terms of $a_2$ directly. This analysis will be undertaken as soon as an oportunity is afforded.—B. Liebowitz.

## ACCELEROMETER TESTING APPARATUS

### Instrument Uses Rotating-Weight Method for Obtaining Harmonic Motion

The obvious way to test an accelerometer is to subject it to a known motion whose acceleration can be calculated and compared with the indication of the instrument. The problem of designing such testing apparatus is therefore that of providing a known cyclical motion, preferably simple harmonic, but not necessarily so.

The principal difficulties involved are to avoid ($a$) distor-

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[4] See THE JOURNAL, November, 1926, p. 433.

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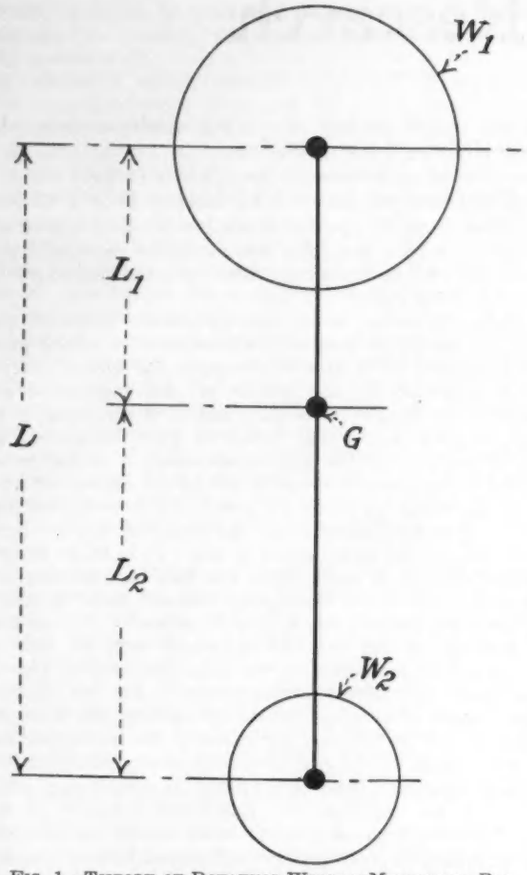


FIG. 1—THEORY OF ROTATING-WEIGHT METHOD OF PRODUCING SIMPLE HARMONIC MOTION

Suppose the System Here Represented, Consisting of Weights $W_1$ and $W_2$ with $L$ as the Center Distance between Them, and $L_1$ and $L_2$ the Distances between $W_1$ and $W_2$ Respectively, and the Center of Gravity at $G$, Rotates around the Center of Gravity $G$; then the Uniform Circular Motion of the Center Point of $W_1$ Can Be Represented by Two Simple Harmonic Motions at Right Angles to Each Other in Space and 90 Deg. Out of Phase. Let a Constraint Be Put on $W_1$ So That It Can Move Only in a Straight Line in the Direction of One of Its Coordinates, $x$; the Circular Motion of $W_1$ Will Be Changed into Its Projection on the $x$ Axis. The Center of Gravity Still Remains at Rest along the $x$ Direction, but It Now Has Simple Harmonic Motion in the Perpendicular Direction

tions and (*b*) parasitic vibrations, to such an extent that the calculated acceleration of the mechanism will agree with its actual acceleration within required limits of error. Unfortunately, there is at present no satisfactory way of checking the performance of such a mechanism itself, except by means of accelerometers, hence the testing equipment must be designed and constructed with proper care. On the other hand, if such apparatus could be made simple and easily reproducible, it would lend encouragement to the further development of accelerometers and would thereby afford material aid to the problem of measuring riding-qualities.

In the spring of 1926, with the cooperation of C. B. Veal and the Overman Cushion Tire Co., we undertook the development of accelerometer testing equipment which aimed to meet the requirements indicated above. The work has suffered many delays, but sufficient data have been obtained to submit the present article as a progress report. In the fall of 1926, Mr. Veal's duties as Research Manager of the Society prevented his taking a further active part in the project, but by that time the most important troubles had been met and dealt with.

## CRANK MOTIONS VERSUS ROTATING WEIGHTS

Automotive engineers would naturally turn to the ordinary crank mechanism or to the scotch yoke to obtain a predetermined cyclical motion. Such linkages, however, have certain drawbacks when it comes to accelerations of high speed and short stroke, which are precisely the accelerations in which the automotive engineer is most interested.

These crank mechanisms each have five bearings; four rotating and one sliding in the ordinary crank, and three rotating and two sliding in the scotch yoke. Furthermore, at least four out of the five bearings are subject to reverse loading. When we bear in mind that at high speeds and short strokes discrepancies of the order of 0.001 in. between the actual and theoretical motions may be significant, the difficulties in applying crank motions to accelerometer testing apparatus become more apparent.

On the other hand, predetermined or at least calculable accelerations can be obtained by unbalanced rotating weights. This method involves only two bearings and these do not have to be fitted with unusual accuracy because of the centrifugal load. This type of apparatus has its own variety of difficulties, of course, but after due consideration we felt that the rotating-weight method was preferable to the crank-motion method on almost all counts, at least insofar as high speeds are concerned.

## THEORY OF THE ROTATING-WEIGHT METHOD

Imagine two weights $W_1$ and $W_2$, connected as shown in Fig. 1, to be rotating with uniform angular velocity and subject to no external forces. According to fundamental principles of mechanics, the center of gravity $G$ of the system will remain at rest. Assume that the weights are symmetrical with reference to a plane passing through $G$ parallel to the paper. Then, if $L$ is the center distance between $W_1$ and $W_2$, the location of $G$ is determined by $W_2 L_2 = W_1 L_1$, where $L_1$ and $L_2$ are the center distances from $G$ to $W_1$ and $W_2$ respectively. Since $L = L_1$ plus $L_2$, we obtain

$$L_1 = W_2 L/(W_1 + W_2) \quad (1)$$
$$L_2 = W_1 L/(W_1 + W_2) \quad (2)$$

Let $\omega$ be the angular velocity. The system is rotating about $G$ with this angular velocity, and the centrifugal forces on $W_2$ and $W_1$ are opposite, and each equal to $\omega^2 W_1 W_2 L/(W_1 + W_2)\, g$. The radial accelerations of $W_2$ and $W_1$ are respectively $L_2\,\omega^2$ and $L_1\,\omega^2$; that is,

$$\text{radial acceleration of } W_2 = W_1 L\,\omega^2/(W_2 + W_1) \quad (3)$$
$$\text{radial acceleration of } W_1 = W_2 L\,\omega^2/(W_2 + W_1) \quad (4)$$

Note that $W_2$ and $W_1$ may each be considered as either rotating or non-rotating about its individual center-of-gravity.

It is well known that the uniform motion of a point in a circle can be represented by two simple harmonic motions at right angles to each other in space and 90 deg. out of phase. That is, if $x$ and $w$ represent the coordinates of the center of $W_1$, then the motion of $W_1$ can be represented by

$$x = W_2 L \sin pt/(W_1 + W_2) \quad (5)$$
$$w = W_2 L \cos pt/(W_1 + W_2) \quad (6)$$

Now, let a constraint be imposed on $W_1$ so that it can move only in a straight line, in the $x$ direction. The effect is to change the circular motion of $W_1$ into its projection on the $x$-axis, so that the displacement of $W_1$ is now given by

$$x = W_2 L \sin pt/(W_1 + W_2),\ w = 0 \quad (7)$$

The center of gravity $G$ still remains at rest along the $x$ direction, but it now has simple harmonic motion in the perpendicular direction given by

$$u = -w = -[W_2 L \cos pt/(W_1 + W_2)] \quad (8)$$

In other words, the effect of imposing the constraint on $W_1$ is simply to impress on $G$ the simple harmonic motion $u$. It is important to note this fact because it eliminates the question whether any distortion will result from the constraint; the angular velocity is not affected by forces acting through the center of gravity.

Due to the constraint on $W_1$, the motion of $W_2$ will no longer be circular either. If $y$ and $z$ be the coordinates of the center of $W_2$, its motion will be given by

$$y = -[W_1 L \sin pt/(W_1 + W_2)] \quad (9)$$

$$z = -[W_1 L \cos pt/(W_1 + W_2)] \\ -[W_2 L \cos pt/(W_1 + W_2)] \\ = -L \cos pt \qquad (10)$$

As is well known, these equations represent an ellipse whose semi-major and semi-minor axes are respectively $L$ and $W_1 L/(W_1 + W_2)$. The centrifugal force on $W_2$, therefore, varies from maximum $W_2 L \omega^2$ to minimum $W_2 W_1 L \omega^2/(W_1 + W_2)$.

These values represent the principal bearing-load, because the weight of $W_2$ will be small compared with centrifugal force in any practical device of this kind. It follows that the contact area between the shaft and bearing will travel around the bearing with $W_2$, hence the bearing will have no tendency to knock even if the clearance is large. In short, the simple arrangement of rotating weights shown in Fig. 1 is theoretically capable of producing pure simple harmonic motion given by

$$x = W_2 L \sin pt/(W_1 + W_2) \qquad (11)$$

whose acceleration is

$$d^2x/dt^2 = -[W_2 L \omega^2 \sin pt/(W_1 + W_2)] \qquad (12)$$

Note that the theoretical amplitude of the motion is the ratio of the unbalanced moment $W_2 L$ to the total weight, the weight of the instrument included. In practice, however, the actual amplitude will differ from this to some extent, because of the frictional and elastic reactions in the supports and attachments. Furthermore, if rotary motions are involved in the supports, corrections must be made for inertia reactions. Hence, the amplitude cannot in general be very accurately predetermined by calculation and is preferably determined by measurement. This adds to the

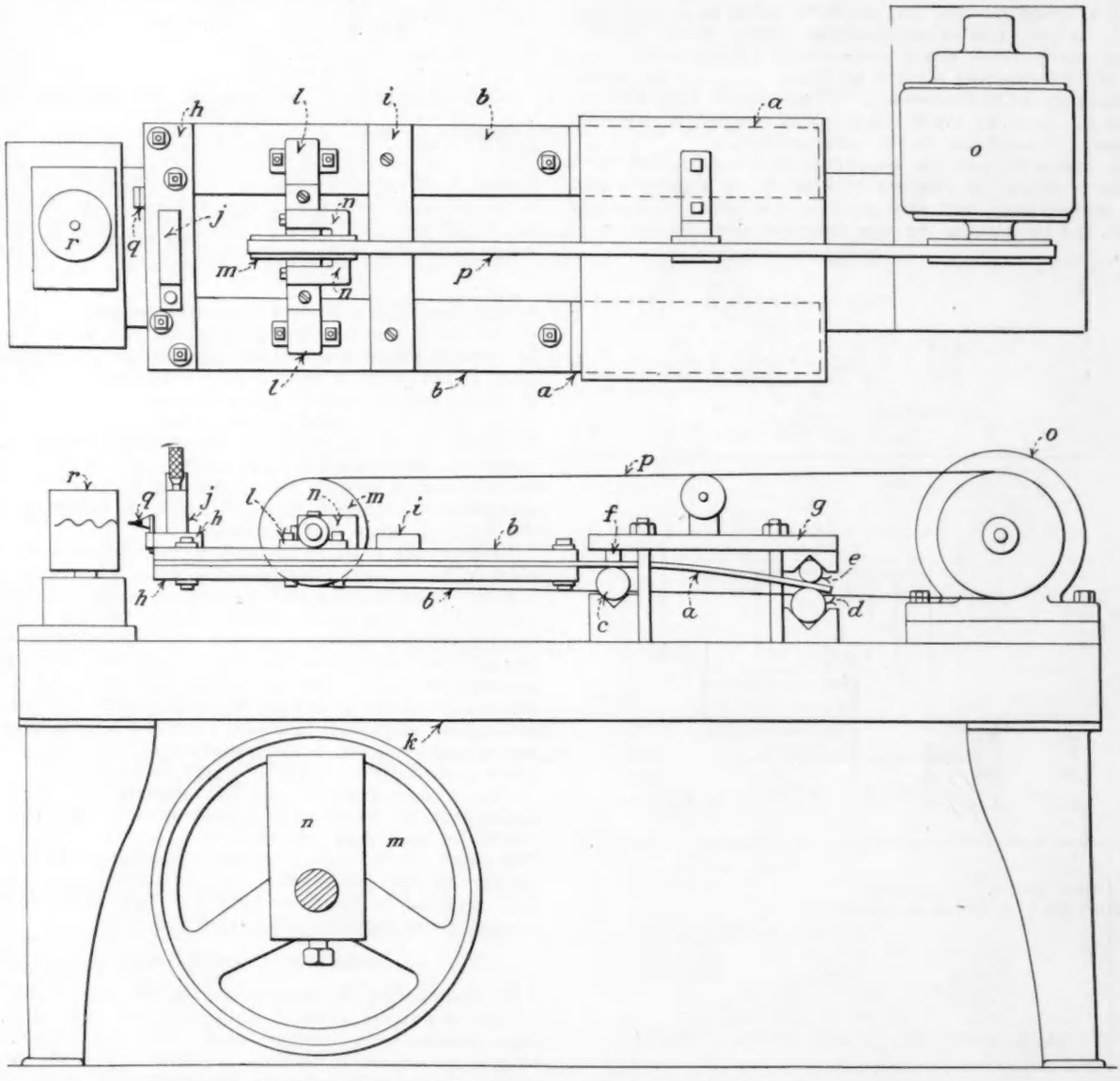


FIG. 2—PLAN AND FRONT ELEVATION OF ACCELEROMETER TESTING APPARATUS

Based upon the Theory Illustrated in Fig. 1, the Instrument Has the Following Parts Corresponding Roughly to the Constrained Weight $W_1$: Four Boards *b*, Two on Either Side of the Apparatus, Each Pair Enclosing a Thin Strip, *a;* Cross-Pieces *h* and *i*, the Former Serving as the Platform Supporting the Accelerometer To Be Tested *j;* and Bearing Blocks *l*. The Rotating System Corresponding to $W_2$ of Fig. 1 Consists of Pulley *m* and Counterweights *n*, Shown Clearly in the Enlarged Detail in the Lower Left Portion of the Drawing, Mounted on Bearing Blocks *l* Driven by Motor *o* by the Belt *p*. The Constraint Imposed upon the Parts Corresponding to $W_1$ Consists of the Two Flat Flexible Wooden Strips *a*, Clamped for Two-Thirds of Their Length between Boards, *b*, and Held in Position by Blocks *c*, *d*, *e*, and *f* and by Plate *g* Bearing on Blocks *e* and *f*. When the Rotating System Is Put into Operation, the Constrained Weight Consisting Roughly of Boards *b* and Everything Fastened Thereto Vibrates with a Simple Harmonic Motion, the Amplitude of Which Is Recorded by Marker *q* on Rotating Drum *r*. The Frequency of the Vibration Is Measured by the Chronograph Arrangement Shown in Fig. 3, Which Is Electrically Connected with the Contacts of Accelerometer *j*. Since the Weight of the System Is Known, the Acceleration of the Vibrations Can Now Be Calculated, and Against This Is Checked the Acceleration as Shown by the Accelerometer. The Entire Apparatus Is Mounted on a Rigid Lathe-Bed *k*

measuring apparatus required but otherwise is no disadvantage.

The simple theory given must be modified to include the reactions of the supports. It is unnecessary to go into detail here further than to state that if the suspension is sufficiently flexible, so that the natural period of vibration of the entire suspended system is long compared with the period of rotation, the actual motion will approximate that prescribed by the simple theory, the degree of approximation depending on the ratio of these periods. If this ratio be very large, the amplitude will remain sensibly constant for small fluctuations in speed.

## Special Design Requirements

The weight $W_1$ of Fig. 1 must include a platform to accommodate the bearings for $W_2$ and the instruments to be tested, and it is essential that this platform move as a whole and that it be free from vibration within itself. Hence, the first design requirements are a *symmetrical* arrangement of unbalanced weights and a *rigid* platform on which to support the bearings and instruments. Furthermore, this platform should be light, to avoid reduction of amplitude, which is inversely proportional to the total weight.

The platform may be supported and constrained in a variety of ways, the simplest of which is by a pivoted arm. This method will not give strictly rectilinear motion, of course, but by making the arm long compared to the stroke

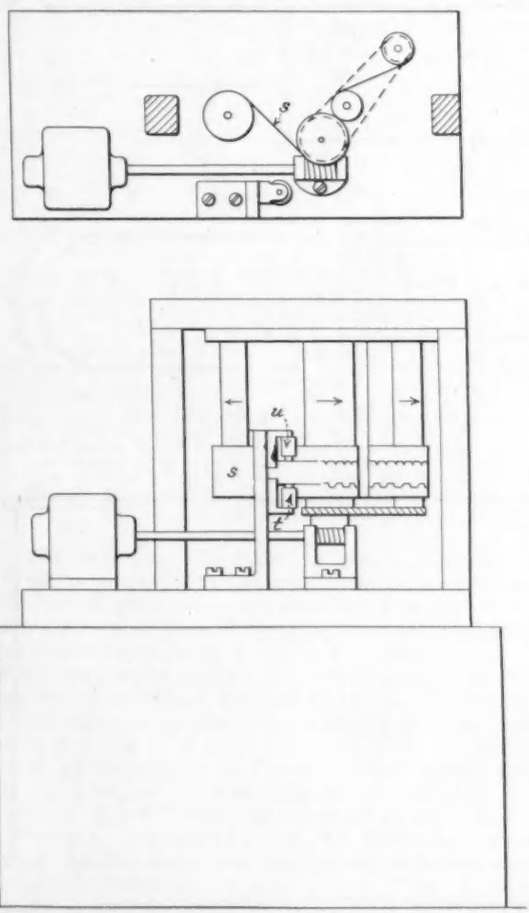


Fig. 3—Chronograph Arrangement for Obtaining a Time Record of Contact Operations

Various Methods of Obtaining a Measurement of Speed Were Tried. The Chronograph Arrangement Shown Was Finally Adopted. This Apparatus Consisted of a Motor-Driven Roller-Mechanism for Driving the Paper Strip *s*, and Two Relays *t* and *u*. Relay *t* Was Connected to a Contact Clock Beating Half-Seconds; Relay *u* Was Connected in the Accelerometer Contact-Circuit and Indicated the Number of Times the Contacts Operated in the Interval Shown by the Contact Clock. Each of the Relays, of Course, Carried a Pointer Which Bore on the Moving Strip of Paper *s*

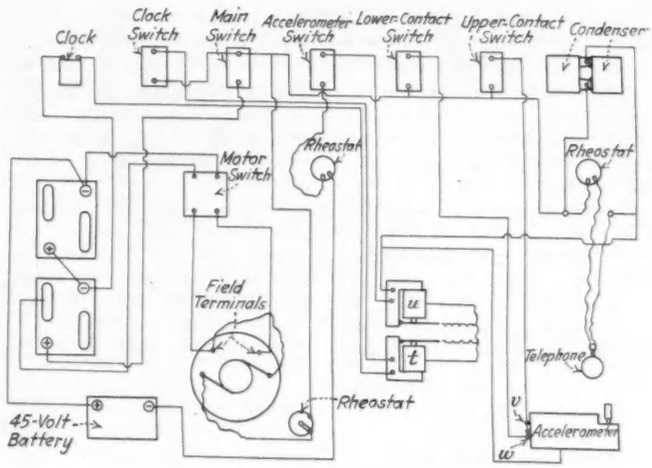


Fig. 4—Electrical Connections for Accelerometer Testing Apparatus

To Operate the Relay *u* at the High Speeds Maintained, a 45-Volt Radio B-Battery in Series with a 12-Volt Storage-Battery Was Used, Giving a Total of 57 Volts on *v*, the Upper, and *w*, the Lower, Contact of the Accelerometer, with a Series Resistance of about 1000 Ohms To Keep the Current Down. In This Way the Effect of the Inductance of the Relay Was Reduced to the Minimum. By the Telephone, Connected with the Contacts of the Accelerometer, the Operator Can Determine when the Latter Is Set so as To Secure an Opening of the Contact with Each Vibration of the Board on Which It Is Mounted

a very close approximation to rectilinear motion is obtained.

The whole arrangement of platform and pivoted arm can be flexibly supported without springs by the simple expedient of balancing it with a counterweight. But on account of the increased inertia and consequent loss of amplitude which this would occasion, and also on account of torque reactions, a spring support was deemed preferable. The design requirements for the spring are that it shall be flexible enough to give the system the required long period and yet stiff enough to avoid free periods of its own within the working range of frequencies.

To drive the rotating system, the best arrangement would seem to be a small electric motor with large bearings and a large shaft having the unbalanced weights fastened directly to it. The motor *could* be small because it has only bearing friction to overcome, and it *should* be small to avoid loss of amplitude. On the other hand, the bearings must be large enough to take the centrifugal load, and the shaft big enough to avoid flexural vibrations in it. In commercial motors the shaft and bearings are too small to handle the centrifugal loads due to the necessarily large counterweights, hence a specially built motor would be required.

Such a special motor was not available for the present experiment, so we resorted to belt drive. The objection, of course, is that parasitic vibrations are introduced by the "flopping" of the belt, especially since the driven pulley travels up and down with the platform. But, since the duty imposed is small, a light belt can be used, thereby keeping these parasitic vibrations small.

## Description of the Apparatus

In constructing the present device, our object was not to produce a finished piece of equipment, but only to assemble from materials on hand or easily obtainable a sufficiently reliable apparatus to indicate whether or not the rotating-weight method is suitable for the purpose in view. We feel that the results have justified our efforts so far and, although the apparatus is relatively crude, it serves to bring out the advantages of the rotating-weight scheme.

In view of the foregoing analysis of the problem, a brief description of the apparatus will suffice. The platform, supporting arm and spring were all combined into a unit as shown in Fig. 2. Two flat wooden strips, *a*, were each clamped between two boards, *b*, leaving about one-third of the length of the thin strips *a* free. These projecting portions of strips *a* serve as the spring, being held in position

by blocks, *c*, *d*, *e* and *f* and by plate *g* bearing on blocks *e* and *f*. Cross-pieces *h* and *i* unite the two side-members, cross-piece *h* serving as the platform supporting the accelerometer shown at *j*. The whole apparatus is mounted on lathe-bed *k*, which serves as a foundation.

Fastened to the side-pieces *b* are the bearing blocks *l* for the shaft which carries the pulley *m* and counterweights *n*. The pulley and counterweights are shown more clearly in the enlarged detail shown as an insert in Fig. 2. Motor *o* drives the pulley *m* by the belt *p*.

To measure the amplitude a marker was fastened to the platform as shown at *q*. A rotating drum, *r*, was brought into contact with this marker, and the amplitude was measured from the resulting curve. Due to the fact that the motion of the arm was a pivotal movement about *f*, the amplitude thus measured had to be corrected because the marker was farther from *f* than the center of the accelerometer. This was a constant correction and was easily applied. The stroke determined in this way varied between 0.166 and 0.175 in., a total variation of 0.009 in., in the entire series of runs to be described.

Our amplitude measuring device was not as satisfactory as might be desired, and the error in this measurement was perhaps the largest single discrepancy.

Various methods of obtaining a measure of speed were tried, and we finally resorted to a chronograph arrangement, shown in Fig. 3. This apparatus consisted of a motor-driven roller mechanism for driving the paper strip *s*, and two relays *t* and *u*. Relay *t* was connected to a contact clock beating half-seconds; relay *u* was connected in the

TABLE 1—SUMMARY OF RESULTS

| 1—Run No. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | |
|---|---|---|---|---|---|---|---|---|
| 2—Speed, r.p.s. | 25.67 | 24.67 | 24.00 | 24.70 | 24.60 | 25.00 | 24.45 | |
| 3—Stroke, in. | 0.1664 | 0.1710 | 0.1690 | 0.1680 | 0.1750 | 0.1660 | 0.1710 | |
| Zero Setting | | | | | | | | |
| 4—Upper Contact | 0.1700 | 0.1697 | 0.1690 | 0.1723 | 0.1722 | 0.1720 | 0.1707 | |
| 5—Lower Contact | 0.1732 | 0.1730 | 0.1733 | 0.1733 | 0.1731 | 0.1732 | 0.1732 | |
| 6—Line 5 — Line 4 | 0.0032 | 0.0033 | 0.0043 | 0.0010 | 0.0009 | 0.0012 | 0.0025 | |
| 7—Average | 0.1716 | 0.1713 | 0.1711 | 0.1728 | 0.1726 | 0.1726 | 0.1719 | |
| 8—Gap, in. | 0.00200 | 0.00210 | 0.00270 | 0.00064 | 0.00058 | 0.00076 | 0.00160 | |
| 9—Theoretical Gap-Error, per cent | 6.05 | 6.07 | 7.00 | 3.40 | 3.10 | 3.70 | 5.10 | |
| Micrometer Setting | | | | | | | | |
| 10—Upper Contact | 0.1300 | 0.1305 | 0.1343 | 0.1352 | 0.1365 | 0.1350 | 0.1345 | |
| 11—Lower Contact | 0.1375 | 0.1390 | 0.1413 | 0.1398 | 0.1397 | 0.1390 | 0.1405 | |
| Acceleration, ft. per sec. per sec. | | | | | | | | |
| 12—Determined from Upper-Contact Micrometer-Setting | 200.0 | 196.0 | 173.5 | 185.5 | 178.5 | 185.0 | 181.0 | |
| 13—Determined from Lower-Contact Micrometer-Setting | 178.5 | 170.0 | 160.0 | 167.5 | 167.0 | 171.0 | 163.5 | |
| 14—Determined from Average-Contact Micrometer-Setting Corrected | 181.5 | 172.0 | 160.5 | 171.0 | 170.0 | 174.0 | 165.5 | |
| 15—Calculated | 180.5 | 171.0 | 160.0 | 168.5 | 173.0 | 171.5 | 168.0 | |
| Deviation from Calculated Value of Acceleration, per cent | | | | | | | | |
| 16—Acceleration Determined from Average Contact-Setting Corrected [100 (Line 14 —Line 15) ÷ Line 15] | +0.55 | +0.58 | +0.31 | +1.46 | —1.76 | +1.43 | —1.51 | Average, 0.12 per cent<br>Average Absolute, 1.11 per cent |
| 17—Determined from Lower-Contact Setting [100 (Line 13 — Line 15) ÷ Line 15] | —1.10 | —0.58 | 0 | —0.60 | —3.60 | —0.30 | —2.71 | Average, 1.27 per cent |
| Deviation of Upper-Contact Instrumental-Determination of Acceleration from Average Instrumental-Determination of Acceleration | | | | | | | | |
| 18—Per Cent, [100 (Line 12 — Line 14) ÷ Line 14] | 10.0 | 14.0 | 8.1 | 8.5 | 5.0 | 6.3 | 9.4 | Average, 8.76 per cent |
| 19—Ft. per Sec. per Sec., Line 12 — Line 14 | 18.5 | 14.0 | 13.0 | 14.5 | 8.50 | 11.0 | 15.5 | |

accelerometer contact-circuit and indicated the number of times the contacts operated in the interval shown by the contact clock. Each of the relays, of course, carried a pointer which bore on the moving paper $s$. The clock was checked after each run by counting the number of beats in 2 min. We believe that our speed determinations are accurate to better than 0.5 per cent.

The diagram of electrical connections is shown in Fig. 4, and requires no comment. It may be pointed out that, to operate the relay $u$ at the speeds at which we ran, we used a 45-volt radio B-battery in series with a 12-volt storage-battery, or a total of 57 volts on the accelerometer contacts $v$ and $w$, with a series resistance of about 1000 ohms to keep the current down. In this way the effect of the inductance of the relay was reduced to the minimum.

Even with this voltage in the accelerometer circuit, the accelerometer setting had to be reduced to operate the relay properly. For this reason, our speed determinations could not be made exactly simultaneously with the accelerometer settings, so that some error crept in due to speed variations. With the apparatus available it was not feasible to adopt the more simple and direct plan of eliminating relay $u$ and having marker $q$ bear directly on the chronograph drum. In later experiments we hope to be able to do this and thereby obtain amplitude and speed on one paper simultaneously with the accelerometer settings.

### Experimental Runs

The accelerometer used in the test is of the micrometer type described in an article entitled Riding-Qualities, published in the Automotive Research department of THE JOURNAL.[6] It was hoped that we would be able to check-up the formula for the gap error when using the lower contact, which was given in an article entitled Contact-Accelerometer Gap-Clearance[7]. When we began to collate the results, however, it became evident that the formula would not apply with sufficient accuracy in the present instance. As stated in the note printed above, this formula was derived on the assumption that the acceleration $a_s$ is sensibly constant throughout the motion of the accelerometer weight from the opening of the upper gap to the closing of the lower gap. While this condition prevails in some types of instrument, the formula does not apply sufficiently closely to the present instrument. The best method under the circumstances is to use the lower-contact zero-setting when using the lower contact, and the upper-contact zero-setting when using the upper contact.

However, a first approximate correction can be obtained by taking $a_s$ from the average value of the upper and lower-contact zero-settings, and applying the above-mentioned formula. This method was also used in working up the results, and gave values in closest agreement with the calculated accelerations, as will be seen from the summary of results given in Table 1.

Owing to the changes which the apparatus was undergoing, systematic runs were not made until very recently, and time has not been available to make many such runs. The experiments which we have tried so far have all been made at practically one speed and stroke but with different gaps.

It will be noted, by comparing lines 13 and 15, and also from line 17, that the lower-contact value of acceleration, using the lower-contact zero-setting, agrees with the calculated acceleration determined from the speed and amplitude measurements within an average error of 1.27 per cent, the largest single discrepancy being 3.60 per cent. It will be noted further that the errors are all minus; that is, that the calculated value is systematically higher than the measured value. If we use the correction mentioned above, we get the results shown in lines 14 and 16, which agree with the calculated values within an average error of 0.12 per cent and average absolute error of 1.11 per cent, the maximum discrepancy being 1.51 per cent. These results on the whole are satisfactory, but too much emphasis must not be placed on the small error obtained from the corrected value, because our runs have not been sufficient in number and the possibility of fortuitous cancellation of systematic errors must not be overlooked. Furthermore, the method of correction applied does not as yet have a sufficiently strong theoretical basis. The uncorrected lower-contact values, however, are sufficiently close to the calculated values to indicate the soundness of the apparatus and the method.

As was anticipated, parasitic vibrations were present, due to the belt and other causes. For this reason we would expect the upper-contact values to be higher than the calculated or the lower-contact values. From lines 12 and 18 it will be noted that the upper-contact values average 8.76 per cent higher than the corrected lower-contact values, the maximum difference being 14.00 per cent, the minimum 5.00 per cent. It should be mentioned that, while the lower-contact settings were fairly sharp, the upper-contact settings were comparatively broad, and some judgment was necessary in making the latter. The influence of the parasitic vibrations is therefore quite marked and prevents any further analysis of the upper-contact values obtained in the present runs. The interesting fact is brought out that the effects of these parasitic vibrations can be successfully eliminated by using the lower contact, at least under the conditions under which these experiments were performed.

In conclusion, we wish to thank H. Centervall for his valuable assistance throughout the work.—C. M. Manly and B. Liebowitz.

[6] See THE JOURNAL, March, 1926, p. 249.

[7] See THE JOURNAL, November, 1926, p. 434.

## RIDING-QUALITIES INSTRUMENTATION

### Accelerometer Developed for Practical Road-Testing Given Extensive Trial

A six-element contact-type accelerometer has been perfected at the laboratories of the Firestone Tire & Rubber Co. and is now in production. In the development of this instrument, gap error and the effect of axle vibrations of short period and high frequency have been eliminated. The instrument is sufficiently rugged to be applied to motor-truck, motorcoach and passenger-car axles. One of these accelerometers has been in such service for several hundred miles, and on its frequent inspections has been found to maintain its calibration within a few per cent. The object in the development work was to design a practical, usable instrument that would give results of sufficient accuracy to be valuable.

This instrument counts the number of times the accelerations occur greater than a predetermined value, hence at the end of a test run it is necessary only to read the counter to get an indication of the average riding-qualities during the test.

Careful investigation has developed the fact that the contacts as installed in the various contact-type accelerometers constructed to date cannot be made to function with a sufficient degree of precision to justify the use of an integrating device. The method of integrating against time the current passing through the accelerometer contacts was abandoned only after numerous road-tests, using a carefully constructed German electrolytic integrator.

The records taken with an oscillograph whose elements were connected in the accelerometer circuit showed conclusively that the contacts could not be made to open and close sharply, that the tendency to chatter was very pronounced and that the time the contacts were closed was as much a function of the contact construction as it was of the acceleration to which the entire instrument was being subjected. Preliminary tests of this new accelerometer have given some very interesting results. These will be published as soon as complete tests have been made.

The possibility of determining riding-qualities in the laboratory is being further investigated. It appears probable that for a certain chassis the vertical axle-movement is an indication of the riding-qualities for that chassis. Immediate future work will consist of an effort to correlate the methods developed and data secured to date.—R. W. Brown.

# Oil-Flow through Crankshaft and Connecting-Rod Bearings

*By* D. B. Brooks[1] and S. W. Sparrow[2]

Semi-Annual Meeting Paper *Illustrated with* Drawings and Charts

THE paper is limited to a discussion of the factors governing flow of the lubricant through the crankshaft and connecting-rod bearings. Apparatus for measuring the flow is described, and the fact that it permits measurement under operating conditions is emphasized. Results obtained by increasing main and connecting-rod bearing clearances are enumerated first. The paper then treats of the influence of engine-speed. Centrifugal force is shown to have a major influence on flow at high speeds, but it is pointed out that the magnitude of this influence can be controlled to a considerable extent by the radial location of the oil-hole in the crankpin.

The fact that the effect of changes in pressure differs with differences in engine-speed and oil-flow is commented upon, and a possible explanation for this condition is advanced.

Some results which at first appeared surprising were obtained in static tests in which the flow was measured at various crank-angles. The paper is concluded with a discussion of the probable explanation of these results and their significance.

TO be satisfactory, a lubricating system must supply the proper quantity of lubricant to the rubbing surfaces. It is with this feature of lubrication, namely, the "transportation" of the lubricant, that this paper deals. If the amount of lubricant furnished is too much or too little, the engine itself is likely to demonstrate that fact in a convincing but often costly way. However, the engine's demonstration does not tell how much is "too much" or "too little." To obtain this information, measurements of flow are essential.

Measurements quoted in this paper were obtained by supplying oil under a known pressure to the bearing or bearings under test and at the same time measuring volumetrically the amount of flow. To accomplish this, the apparatus shown in Fig. 1 was employed. This consisted of a tank constructed of 3 ft. of 4-in. pipe capped at each end and supplied with a gage-glass, filling pipe, pressure-gage, and suitable valves and piping for controlling the pressure of the compressed air which served to force the oil through ¼-in. tubing to the engine under test. At the engine, a pressure-gage was attached to the feed-line to give the actual delivery pressure of the oil. When the oil-flow to but one bearing was being studied, the other bearings were fed in the normal manner, the oil-distributor pipe being blanked-off to the bearing under test. When the flow to all bearings was being measured the oil-pump was disconnected and the bearings were supplied directly from the metering tank. The gage-glass of the metering tank was graduated into equal calibrated sections, and measurements of oil-flow were made by noting with a stop-watch the time required for the oil-level in the gage-glass to fall past one or more of these graduations.

Preliminary tests showed that the oil-flow was the same when the engine was being driven by the dynamometer as when it was operated under its own power. For convenience, therefore, most of the measurements reported herein were obtained under the former conditions. Several series of measurements were also obtained with the engine stationary and the crankshaft in various angular positions. These measurements were made for the purpose of ascertaining the distribution of flow over the cycle and are referred to as "static" measurements.

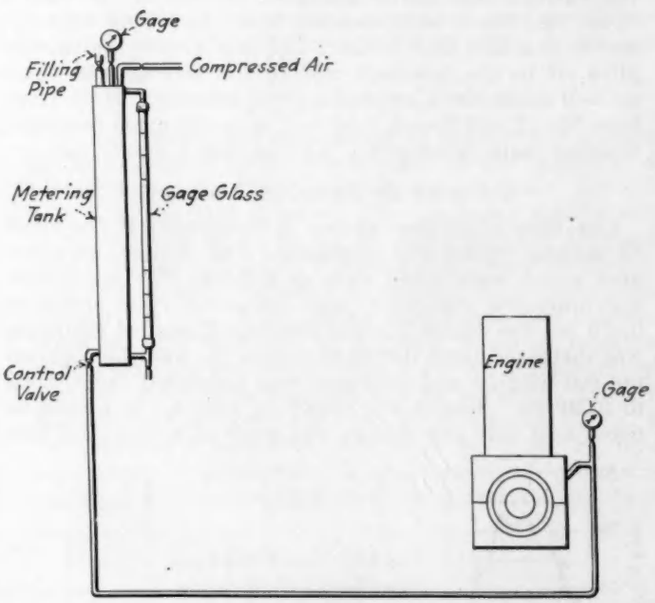


Fig. 1—Apparatus Used for Volumetric Measurement of Oil-Flow through Bearings

The temperature of the oil in the engine and in the metering tank was brought to 70 deg. fahr. at the start of the test, and the temperature of the jacket water at the outlet was maintained at that value until the test was completed. Sufficient time was allowed to elapse between successive tests to permit the engine parts to cool to a temperature of approximately 100 deg. fahr.

The original purpose of this work was to find the relative amounts of oil supplied to the four bearings of an experimental engine. This information was obtained and, as a result, the quantity of oil fed to one of these bearings was increased. Many of the data are of interest only in connection with the particular engine tested, but some of the data are of general interest and

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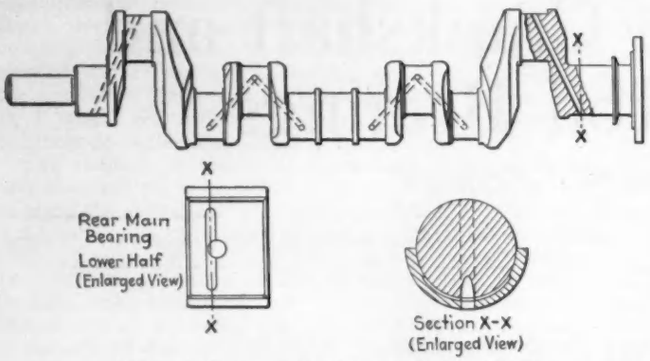


FIG. 2—CRANKSHAFT OF THE ENGINE USED
Locations of the Oil-Supply Holes and of the Oil-Groove of the Rear Main-Bearing Are Shown. Cross-Sections through the Crankshaft at *XX* Are Shown Also

it is this latter portion that is to be discussed. Before presenting this discussion, however, attention is called to the lubricating system of the engine. A drawing of the crankshaft is shown in Fig. 2. Oil is drawn from the sump to the pump and is then forced through an oil-distributing manifold to the four main bearings. The connecting-rod bearings are fed from the drilled crankshaft, holes in the main crankshaft-journals registering with a circumferential groove in the main bearings for a portion of each revolution. The amount of registry of the various bearings is indicated in Fig. 3. It will be noted that No. 1 main bearing has a very long registry, due to the fact that it has a 210-deg. groove which supplies oil to the camshaft and to the timing-gear chain as well as to No. 1 connecting-rod bearing. Main bearings Nos. 2 and 3 each feed two connecting-rod bearings, whereas main bearing No. 4 feeds only a single rod.

## CHANGES IN BEARING CLEARANCE

One item of interest shown in these tests is the effect of changes in bearing clearance. The changes in clearance which were made were as follows: The main-bearing diametral clearance was increased from 0.005 to 0.010 in., the connecting-rod bearing diametral clearance was increased from 0.0030 to 0.0055 in., and the connecting-rod-bearing end-clearance was increased from 0.008 to 0.020 in. Results are shown in Fig. 4. It should be noted that only one change was made at a time and that

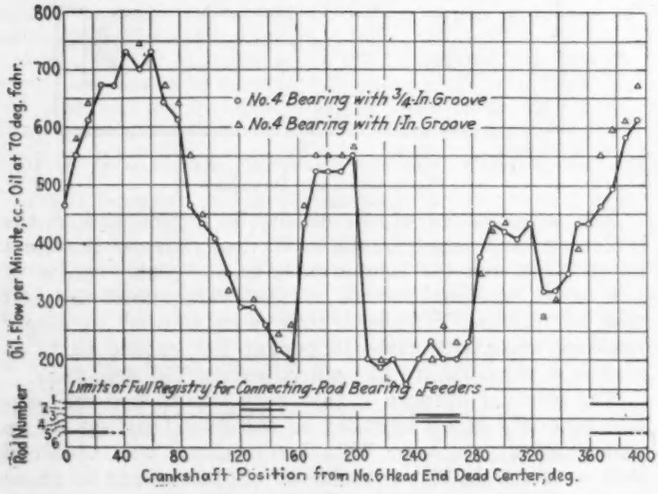


FIG. 3—STATIC OIL-FLOW FROM ALL BEARINGS
The Chart Shows the Rate of Oil-Flow, Measured at Different Crank-Angles throughout the Cycle, and Also the Limits of Registry of the Connecting-Rod-Bearing Oil-Supply Holes

the original clearance was then restored before another change was made. Hence, in Fig. 4, the difference between the upper and lower curves represents the increase in flow due solely to the increase in main-bearing clearance, and the difference between the lower curve and that immediately above it represents the increase in flow due solely to the increase in connecting-rod-bearing diametral-clearance.

In this case, at least, increasing the connecting-rod-bearing end-clearance had practically no effect upon oil-flow. This is not surprising considering that the original end-clearance of 0.008 in. was much greater than the diametral clearance of 0.003 in. It will be observed also that, in this particular engine, the increase in main-bearing clearance increased the flow much more than did the increase in connecting-rod-bearing clearance. One must remember, however, that the main bearing is subjected to the supply pressure continuously, whereas the

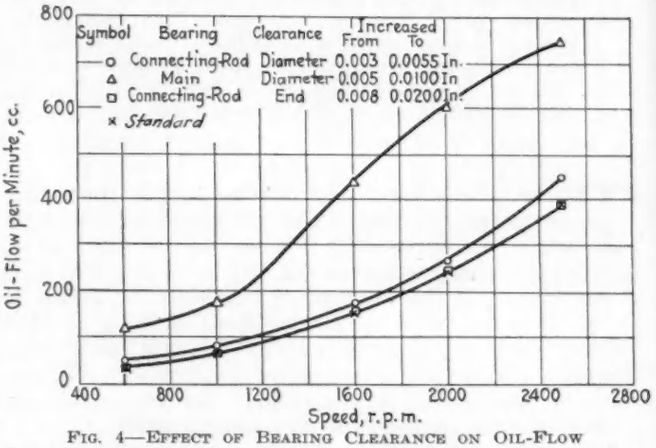


FIG. 4—EFFECT OF BEARING CLEARANCE ON OIL-FLOW
The Data Are for No. 4 Main Bearing of Engine 8F-109, the Delivery Pressure Being 57 Lb. per Sq. In.

connecting-rod bearing is subjected to this pressure for only a small portion of each revolution. The connecting-rod bearing is, however, subjected continuously to the centrifugal pressure developed by the column of oil in the crank-arm. This pressure, as will be shown later, is fairly high at high speeds. With high speeds and low pressures applied to the line feeding the main bearing, the relative effect of increasing main-bearing and connecting-rod-bearing clearances might be considerably different.

## EFFECT OF SPEED

In considering the effect of speed upon rate of oil-flow to connecting-rod bearings, it must be borne in mind that the total time during which the oil-groove in the main bearing registers with the hole through which oil is supplied to the connecting-rod is independent of speed. For example, if engine-speed is doubled, there will be twice as many registries per minute, but the duration of each registry will be one-half as long. Nor should one forget that measurements of oil-flow include the flow to both main and connecting-rod bearings. There is, however, considerable justification for the belief that the major portion of the increase in flow with increase in speed represents increased flow through connecting-rod bearings.

Another fact not to be overlooked is that, under the conditions of this test, the pressure forcing oil to the bearings was the same at all speeds. This is the condition which prevails in engines that have a pressure-relief valve of adequate capacity opening at a low engine-speed. In the absence of such a valve, the ordinary

gear-pump would give a supply pressure which would increase with speed. This would mean that the effect of speed would be even greater than is shown in Fig. 5.

The flow through No. 4 main bearing, as shown in Fig. 5, is of greatest interest as this bearing supplies only one connecting-rod and a large proportion of the increase in flow with increase in speed would be expected, therefore, to be due to the flow through the connecting-rod. The flow to No. 4 main bearing, when re-plotted on a logarithmic scale, shows that the flow above a small constant value of about 18 cc. per min. varies as the 2.4 power of the speed.

If flow to the connecting-rod bearing were truly viscous and depended only on centrifugal pressure, the flow would vary as the square of the speed. Calculation of the pressure due to centrifugal force is given in the Appendix to this paper. It is there shown that, for this crankshaft, the centrifugal pressure at the surface of the connecting-rod bearing is $0.00000486\ N^2$, where $N$ is the number of revolutions per minute. One would hardly expect that the flow through the connecting-rod bearing would be in exact agreement with this square relationship, however, as a change in speed changes not only the temperatures of the shaft and the bearing but the crank angles at which the inertia forces become greater or less than explosion and compression pressures.

Fig. 6 shows a comparison between the flow with the standard crankshaft and with a shaft having its oil-supply hole drilled to the inner side of the crankpin as shown in Fig. 7. The flow with this shaft increased with the 2.9 power of the speed but, as will be evident

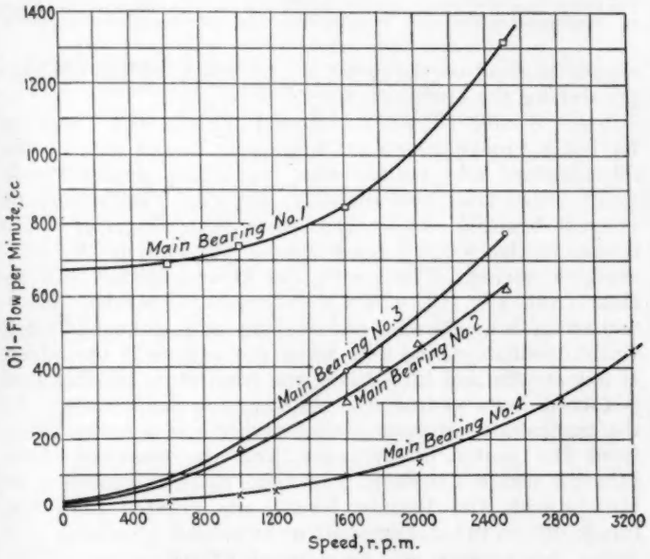


FIG. 5—COMPARISON OF BEARINGS
The Oil-Flow from Each Bearing of Engine 8F-109, Plotted against Speed, Is Indicated, the Delivery Pressure Being 57 Lb. per Sq. In.

from Fig. 6, was much less at the higher speeds than that obtained with the standard shaft. The oil-flow values which are plotted represent total flow to main and connecting-rod bearings, but, inasmuch as the change in shafts should not have affected the flow through the main bearing, the difference represents the difference in flow to the connecting-rod bearing. That it is reasonable to expect a large difference in centrifugal force will be evident from a comparison of the calculated values given in Table 1.

From a consideration of the two curves in Fig. 6 it will be evident that, by merely changing the location of the oil-discharge hole, a wide variation in the amount of

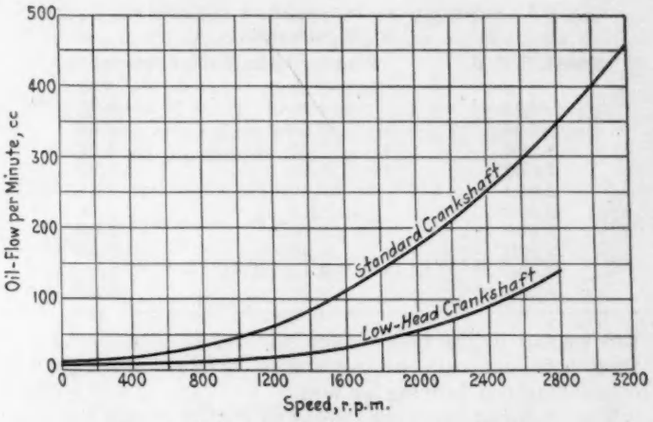


FIG. 6—VARIATION OF OIL-FLOW WITH SPEED
The Oil-Flow to Rear Main-Bearing No. 4 and to No. 6 Connecting-Rod Bearing Is Indicated

oil discharged at high speeds can be obtained. Whether such a change will affect the efficacy with which the oil is spread over the bearing should, of course, be given consideration. By varying oil-pump pressure in conjunction with the radial location of the discharge hole to the connecting-rod bearing, almost any relation between speed and oil-flow should be obtainable.

In connection with the very considerable influence of centrifugal force upon oil-flow, attention is called to the fact that a change in engine stroke might change flow characteristics to a marked extent, as the centrifugal force depends upon the crank radius, which is one-half the stroke.

### FLOW IN RELATION TO PRESSURE

At low speeds, and particularly at low rates of oil-flow, the flow appears to be proportional to the pressure; at higher flow-rates, this is not always the case. Thus, in Fig. 8, at the highest speed, flow increases rapidly with pressure up to 60 lb. per sq. in. and thereafter increases only about as rapidly as it does at the lower speeds.

A theory which appears to explain these facts satisfactorily is that, with higher speeds and low delivery-pressures, the tendency is to throw more oil from the drilled hole in the crank-cheek into the connecting-rod bearing than can be replaced during registry. Thus, in the case of the 2800-r.p.m. run illustrated in Fig. 8, as the pressure is increased from 30 to 60 lb. per sq. in. it acts simply to force more oil into the drilled hole during registry, while above that pressure it replaces all the oil

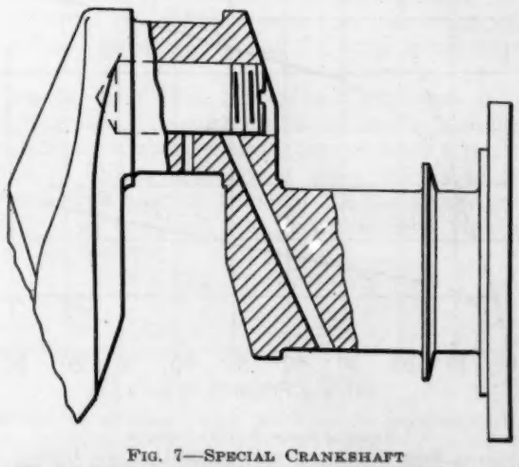

FIG. 7—SPECIAL CRANKSHAFT
The Special Low-Head Crankshaft Shown Is Sectioned To Show the Shape of the Oil-Supply Hole

TABLE 1—CENTRIFUGAL PRESSURE IN POUNDS PER SQUARE INCH ABSOLUTE

| Speed, R.P.M. | Standard Shaft | Special Shaft |
|---|---|---|
| 0 | 0.00 | 0.00 |
| 400 | 0.78 | 0.12 |
| 800 | 3.06 | 0.46 |
| 1,200 | 7.00 | 1.04 |
| 1,600 | 12.50 | 1.85 |
| 2,000 | 19.50 | 2.88 |
| 2,400 | 28.10 | 4.16 |
| 2,800 | 38.20 | 5.60 |
| 3,200 | 49.80 | 7.40 |
| 3,600 | 63.10 | 9.30 |
| 4,000 | 77.80 | 11.50 |

thrown out of the hole before registry is terminated and then acts against the resistance to flow through the connecting-rod bearing as well.

The curve at the upper right of Fig. 9 shows the non-registry flow which would occur if the connecting-rod-bearing resistance were nil and if the drilled hole could replace completely its oil supply during registry. Since the observed total flow at speeds of more than 1800 r.p.m. is but a small fraction of this amount, it is evident that the centrifugal pressure at the connecting-rod bearing is practically unchanged during the non-registry part of the revolution. On this assumption, and considering the resistance to flow of the connecting-rod bearing to be the same as that obtained by static measurements, the curve at the lower right of Fig. 9 was computed. This curve gives the probable non-registry flow through the connecting-rod bearing due to centrifugal pressure. It will be observed that this curve is substantially parallel with the lowest of the observation curves from 2000 to 2800 r.p.m. This fact constitutes a further check upon the theory stated previously.

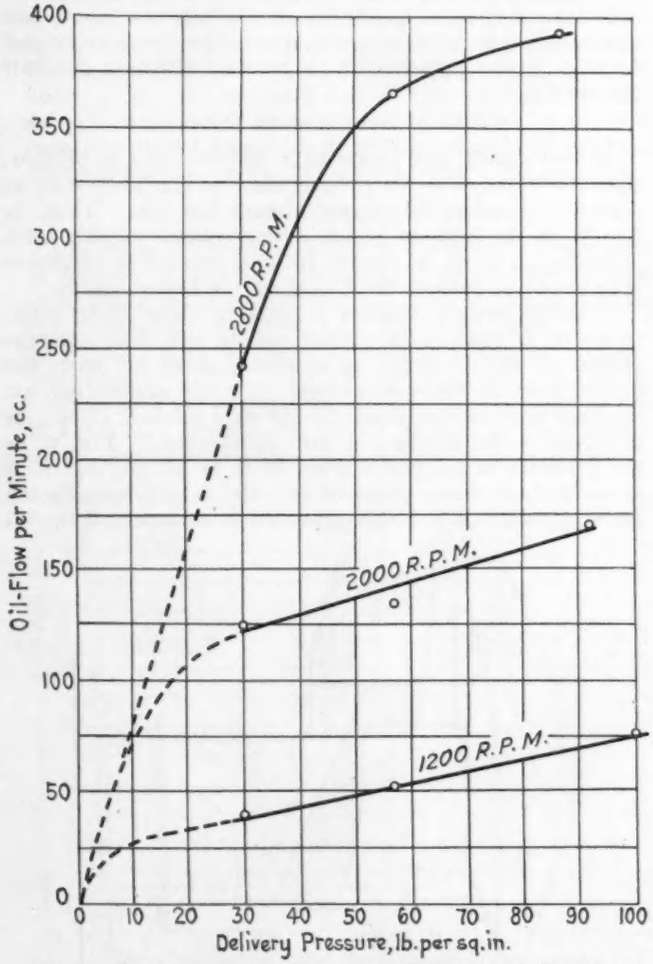


FIG. 8—VARIATION OF OIL-FLOW WITH PRESSURE AT DIFFERENT SPEEDS, MAIN BEARING NO. 4

It Is Evident from the Curves Shown That, at the Highest Speed, Flow Increases Rapidly up to 60 Lb. per Sq. In. and Thereafter Only about as Rapidly as at the Lower Speeds

Reference already has been made to the "static" measurements of oil-flow which were obtained at various crank-angles with the engine stationary. Fig. 3 shows the variation in flow over a complete cycle, while Fig. 10 shows the flow under similar conditions through No. 4 main bearing only. Fig. 11 shows the position of the crankpin and the bearing at top dead-center, the clearance being exaggerated to show the sealing of the oil

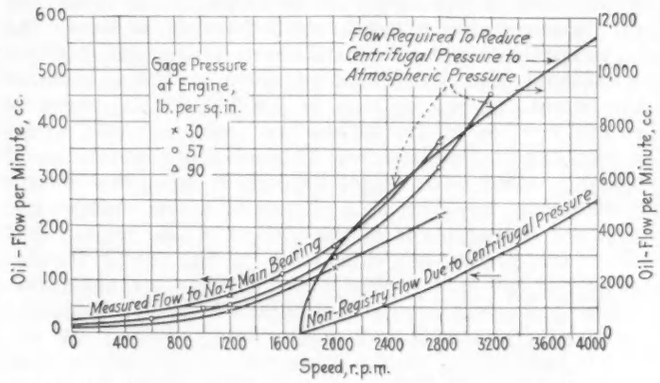


FIG. 9—EFFECT OF PRESSURE ON RATE OF OIL-FLOW TO NO. 6 CONNECTING-ROD BEARING ON ENGINE 8F-109

The Chart Shows Variation of Oil-Flow with Speed and with Pressure, the Oil-Flow Required To Reduce Centrifugal Pressure to Atmospheric Pressure and the Probable Non-Registry Oil-Flow

supply at dead-center and the restriction of the oil supply during the entire registry.

Figs. 3 and 10 were obtained with a rear bearing having a groove which extended only to one side of the oil-entrance hole rather than with the double-length groove used later and shown in Fig. 2. The feature of interest brought out by Fig. 10 is that no material increase in flow occurs until almost the end of the full-registry period. The reason for this condition will be clear from Fig. 12, which shows an exaggerated cross-section of a connecting-rod bearing and journal. Under static conditions, and also when the engine is operating at low speeds and high loads, the connecting-rod bearing is forced down against the shaft at top dead-center. As the crankshaft revolves, the oil-supply hole *a* moves away from the center of pressure, and the clearance area through which discharge can take place increases. At high speeds the inertia forces are greater than the forces due to the compression or explosion pressures, and upper dead-center will be a point of maximum rather than minimum clearance. With the oil fed to the inside of the crankpin, as shown in Fig. 7, the reverse condition is true.

The condition just discussed is probably of minor importance, for the chief function of registry is not to feed the connecting-rod bearing but to feed the oil-hole in the crankshaft which, in turn, feeds the connecting-rod bearing throughout the cycle. Under certain circumstances, however, the flow during registry might be extremely important, and for that reason the results obtained in these "static" tests have been commented upon at some length.

That the information presented is fragmentary, is realized full well. It is believed, however, that it furnishes data not readily available elsewhere relative to

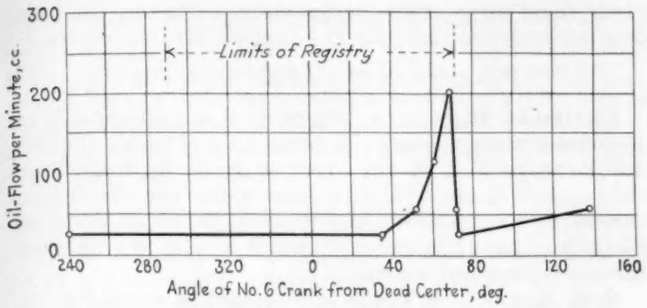


FIG. 10—STATIC OIL-FLOW
The Oil-Flow to Rear Main-Bearing No. 4 and to the Connecting-Rod Bearings Is Indicated for Various Crank-Angles for Engine 8F-109, Which Is Equipped with a Standard Crankshaft

the influence of speed and pressure upon oil-flow, and that it also presents a satisfactory method for measuring oil-flow to an individual bearing under operating conditions.

## APPENDIX

### CALCULATION OF CENTRIFUGAL PRESSURE ON OIL IN THE CRANK-CHEEK

Let

$A$ = Area of the drilled hole in the crank-cheek, equal to 0.0491 sq. in.
$D$ = Density of oil, equal to 0.0325 lb. per cu. in.
$F$ = Centrifugal force on the oil in the crank-cheek
$g$ = 32.16 ft. per sec. per sec.
$l$ = Perpendicular distance from the axis of the crankshaft to the surface of the oil in the drilled hole in the crank-cheek. See Fig. 12
$L$ = Perpendicular distance from the axis of the crankshaft to the surface of the connecting-rod bearing, equal to 0.271 ft. See Fig. 12
$N$ = Number of revolutions per minute
$P$ = Centrifugal pressure at the surface of the connecting-rod bearing, in pounds per square inch

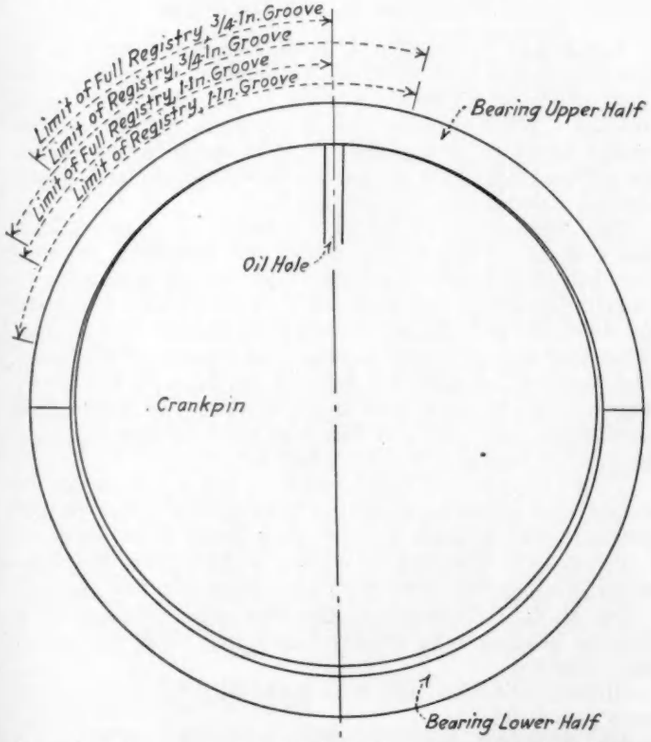


FIG. 11—POSITION OF CRANKPIN AND BEARING AT TOP DEAD-CENTER
The Amount of Clearance Is Exaggerated To Show the Sealing of the Oil Supply at Dead-Center and the Restriction of the Oil Supply during the Entire Registry

$\Theta$ = Angle between the axis of the drilled hole in the crank-cheek and a perpendicular to the axis of the crankshaft
$V$ = Oil-flow in cubic centimeters per minute
$W$ = Weight of oil in the crank-cheek between the axis of the crankshaft and the surface of the connecting-rod bearing, in pounds

Then, from the fundamental formula

$$F = W\omega^2 r/g \tag{1}$$

we have

$$AP \sec\Theta = [W/g]\,[(2_\pi N^2/60)]\,[(L + l)/2] \tag{2}$$

Since

$$W = 12\,DA\,(L - l)\sec\Theta \tag{3}$$
$$P = [12\,D/g]\,[(2\pi N^2)/60]\,[(L^2 - l^2)/2] \tag{4}$$
$$= 0.0000664\,N^2\,(L^2 - l^2) \tag{5}$$

When $l = 0$

$$P = 0.0000664\,N^2L^2 \tag{6}$$

and, in the case under consideration, as $L = 0.271$,

$$P = 0.00000486\,N^2 \tag{7}$$

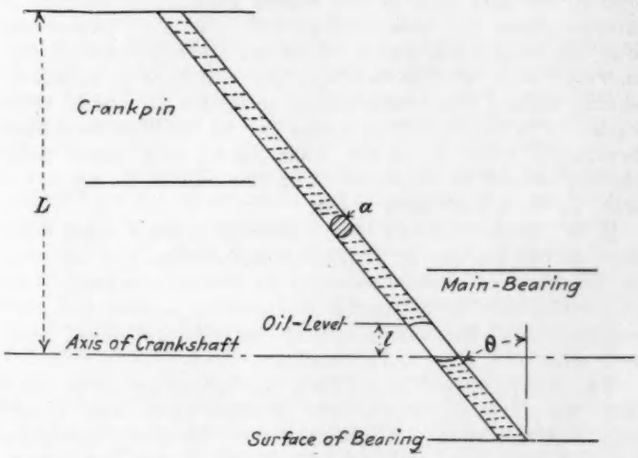


FIG. 12—EXAGGERATED CROSS-SECTION OF A CONNECTING-ROD BEARING AND JOURNAL
The Drawing Illustrates the Calculation of Centrifugal Pressure

If a perfect seal is obtained, that is, if no air gets into the drilled oil-hole, then a back-pressure is developed, due to the column of oil between the axis of the crankshaft and the main-bearing surface, which can be calculated from equation (6). However, if air does get into this hole, as seems probable, the expansion of the air nearly annuls the back-pressure, which, however, is still active in retarding oil-flow during registry. In this latter case the developed pressure at the crankpin due to centrifugal force approaches the value given by equation (6).

At speeds such that centrifugal pressure is great enough to create a vacuum in the drilled hole and cause non-registry flow with resultant loss of head $l$, the necessary flow to reduce pressure at the crankpin to atmospheric pressure, that is, to equal the maximum possible flow with a perfect seal, can be determined from equation (4), as follows:

$$l^2 = L^2 - (10^7\,P/664\,N^2) \tag{8}$$

Since $P = 14.7$

$$l = (L/N)\,\sqrt{[N^2 - (3.02 \times 10^6)]} \tag{9}$$

and

$$V = 16.4 \times 12\,l\,AN \sec\Theta \tag{10}$$
$$= 3.13\,\sqrt{[N^2 - (3.02 \times 10^6)]} \tag{11}$$

Also, from equation (9), $l = 0$ when $N = 1738$ r.p.m., the speed at which centrifugal pressure at the crankpin equals atmospheric pressure.

## THE DISCUSSION

D. P. BARNARD 4TH[3]:—Has Mr. Brooks made any attempt to correlate his observations by the method we used several years ago? We tried a simple bearing running without any of the complications of a bearing in an internal-combustion engine. We found that we could regard that bearing as a pump, the flow through which was governed by the pressure developed in the oil-film itself, to which was superimposed any oil-feed pressure which might happen to be in the line. That was a very convenient way of correlating our data.

D. B. BROOKS:—We have not been able to do that because our measurements include the flow to both the main and the connecting-rod bearings. Under operating conditions, it would be very difficult to separate the relative amounts of flow through the main and the connecting-rod bearings. One or two points seem to be contrary to general theory. The two curves in Fig. 6 show no point of change in curvature, which would be expected, due to the fact that at the higher speeds the inertia is greater than the compression and explosion pressures. For the conditions under which we were operating, the inertia force at top dead-center, which was greatest, should exceed the compression pressure at about 1600 r.p.m. The center of pressure in the oil-film therefore begins to shift at about 1600 r.p.m. and some such change would be expected, but the curve is perfectly true up to about 3000 r.p.m.

S. W. SPARROW:—In that connection, some work was done at the Bureau of Standards some time ago relative to the study of oil-flow through an aviation engine. One of the elements investigated consisted of a plain journal-bearing. My observation is that the results checked very well with Mr. Barnard's results.

MR. BARNARD:—What called my attention to it was that the formula which Mr. Brooks gave was about what one might expect within reason from the methods we used some time ago; that is, if the load is considered as increasing as the square of the speed, the pumping tendency increasing directly as the speed, and the volumetric efficiency of the pump dropping off somewhat with the speed, the oil-flow should be proportionate to something between the second or third power of the speed.

CHAIRMAN H. L. HORNING[4]:—Does not the paper represent a report of progress rather than an attempt to draw any too certain conclusions?

MR. BROOKS:—The paper was formulated simply to give such information as we had obtained with regard to the effect of speed and pressure on oil-flow, and to describe the method which gave satisfactory results in measuring the oil-flow to an individual bearing. The formula for pressure at the surface of the crankpin, due to centrifugal force, is computed purely on a mathematical basis; it is not an empirical formula, but the formulas for variation of oil-flow with speed are empirical.

MR. SPARROW:—The comparison was shown with the high-head and low-head crankshaft. The difference as shown is the actual difference which must be due to the centrifugal force of the action alone, so far as the bearing is concerned, because the bearing was the same.

### FLOODING OF CYLINDERS WITH OIL

CHAIRMAN HORNING:—Where does the oil come from that floods the cylinder? A general belief exists that the crank-cheeks pick up the overflow from the main bearings and let it out. In some cases under our observation something like three or four times as much oil from the main bearings was thrown into the cylinders as from the connecting-rod bearings.

E. S. MARKS[5]:—We have had success in putting a hole in the connecting-rod itself. We know where a large amount of the oil on the cylinder-wall comes from.

CHAIRMAN HORNING:—On which side of the connecting-rod is the hole?

MR. MARKS:—It is on the left side when looking toward the engine from the front.

CHAIRMAN HORNING:—Then it is on the trailing side.

RALPH R. TEETOR[6]:—In an experiment we made in connection with oil consumption past the piston-rings in a well-known engine we plugged the holes in the connecting-rod and could detect no difference in the oil consumption in the combustion-chamber; so we were unable to determine whether the holes delivered any more oil to the cylinders. According to the quantity of oil that passed the piston-rings, there was no difference with holes or without. The holes were drilled on the same side as mentioned by the previous speaker.

LEE W. OLDFIELD[7]:—I have been experimenting with a popular make of engine that has holes drilled in both sides of the connecting-rod. I have increased them from 1/16 to 5/32-in. diameter without any apparent difference in either the oil consumption or the operation of the engine. I do not find that the holes do much good. I have not plugged the holes yet but am on the point of doing so.

### PRESSURES IN THE OIL-FILM

CHAIRMAN HORNING:—I have seen it stated that, at certain speeds and with certain viscous oils, pressures amounting to 10,000 lb. per sq. in. are developed in an oil-film. What information is available on the way in which pressure accumulates in that wedge which occurs in all bearings, that is, on the eccentric surfaces made by the journal and its bearings?

MR. BARNARD:—In one of the bulletins of the Kingsbury Machine Works, describing its bearings, mention is made of the results of some tests on an experimental bearing in which the bearings carried a load of upward of 8000 lb. per sq. in. without rupturing the oil-film. That load was sufficient to cause the babbitt of the bearing shoes to squeeze out beyond the edge of the shoe; and yet at no time was there evidence of scoring or scratching nor did the bearing performance indicate that there was any metal contact whatever.

Under the average load of 8000 lb. per sq. in., no doubt exists that somewhere in that bearing-block the oil-film pressure was at least two or three times that intensity.

CHAIRMAN HORNING:—Where wedges like that develop in a bearing some very high pressures are natural.

DR. H. C. DICKINSON[8]:—The maximum pressure in a journal bearing may readily reach five times the average pressure.

QUESTION:—Were any runs made with the connecting-rods removed?

MR. BROOKS:—No.

CHAIRMAN HORNING:—What would result from blocking-off the holes?

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[3] M.S.A.E.—Research engineer in charge of dynamometer laboratory, Standard Oil Co. of Indiana, Whiting, Ind.

[4] M.S.A.E.—President and general manager, Waukesha Motor Co., Waukesha, Wis.

[5] M.S.A.E.—Chief engineer, H. H. Franklin Mfg. Co., Syracuse, N. Y.

[6] M.S.A.E.—Chief engineer, Perfect Circle Co., Hagerstown, Ind.

[7] M.S.A.E.—Consulting engineer in charge of design and development, Pac-Age-Kar Corporation, Chicago.

[8] M.S.A.E.—Physicist and chief of division of heat and power, Bureau of Standards, City of Washington.

MR. SPARROW:—If the holes were blocked-off, the result probably would be fairly comparable with those that Mr. Barnard reported several years ago. This particular study was undertaken primarily to find out why one connecting-rod bearing did not get as much oil as we thought it should get, and did not include the study of the main bearing by itself.

### EFFECTS OF OIL-GROOVES

CHAIRMAN HORNING:—What has been the experience with no grooves in the connecting-rod bearing?

MR. SPARROW:—Nothing was done in this test with reference to oil-grooves. Some experiments were made at the Bureau of Standards with bearings in which the presence or absence of the lubricating film was shown by an electric light, the circuit of which was broken by the film. That is a more or less qualitative method but, under certain conditions, there was a continuous light and, under other conditions, no light and a considerable region in between where intermittent flashes occurred. A number of bearings were tested under such conditions as regards speed and load that there was no light, proving that there was a complete oil-film. Each of those bearings was then grooved in one way or another and, under those same conditions, they showed continuous light, indicating bearing all the time and proving that the groove did destroy the oil-film. In making a statement of this kind, however, the fact should not be overlooked that there may be a need for some type of groove to take care of the dirt which comes through the bearing; that is, something of the kind may be needed to permit a sort of flushing action. With a crankpin which is drilled with a large axial hole and a small hole at right angles to it for feeding the connecting-rod bearing, I think everyone has had the experience that when the plugs are taken out from the end a considerable accumulation of foreign matter is found. Some of the dirt must go through the bearing, and it is entirely reasonable that some channel to take this dirt out directly would be advantageous.

CHAIRMAN HORNING:—It is important to note that, without oil-grooves, it is possible to get complete lubrication in these tests and, under the same conditions with oil-grooves installed, the lubrication is intermittent. The other point Mr. Sparrow makes is that the groove may afford great advantage in getting dirt out. We had trouble with a camshaft front-bearing having 0.0015-in. clearance. Remembering the direction of flow as indicated in the glass journal with which Mr. Barnard experimented, we made a groove and then were able to feed dirt into that bearing and throw it out again without any difficulty.

MR. BARNARD:—Occasionally a bearing in an internal-combustion engine depends entirely for its oil supply on the oil-flow from the supply tube and does not receive oil from the ends of the bearing. Consider, for instance, a bearing 2 in. in diameter. There is about 1 ft. of clearance space open at the ends for oil to get out. If no oil-groove is present, there may be only about ¾ in. or possibly 1 in. of clearance space at the perimeter of the oil-hole for oil to get in. We think it may be better to put an oil-groove in a bearing to supply area for oil entry which is at least comparable to the effective area available for oil escape, on the theory that it is better to build-up an oil-film and destroy part of it than not to supply sufficient oil to build-up the film at all.

CHAIRMAN HORNING:—Has anyone a practical suggestion as to a successful attempt in using a bearing that has no oil-groove?

A MEMBER:—The Rickenbacker Big 8 has a groove milled in the shaft. There is a longitudinal groove along the crankpins which acts as a spreader to get the oil out along the bearing. An end-mill cut is taken along the crankpins such that a section of the pins becomes a circle with a portion of a segment removed. The intersection of the chord of the segmental portion removed and the arc representing the bearing surface is on the trailing side, as I recall.

W. R. STRICKLAND[9]:—Discussion with regard to plain bearings should carry a factor for production, especially on new jobs. It is difficult to get the oil into the bearings when first fitted-up if no oil-grooves are provided. Further, after an engine has stood a long time the oil runs out of the bearings and, when the engine is started, an immediate supply of oil to the bearings is needed; oil-grooves are often installed for this purpose that are not necessary after the engine gets under way and the regular supply of oil reaches them.

CHAIRMAN HORNING:—During the war when we were designing the Class-B motor-truck engine we decided to use a force-feed system of lubrication and then the question came up as to what oil pressure should be used. Pressures from ½ lb. up to 100 lb. per sq. in. were advocated and therefore we thought there must be some common factor. We came to the conclusion that no matter what their oil pressures were, all the engines using force-feed lubrication required about the same amount of oil per minute; in other words, the common factor was about 2½ gal. per min. past the bearings in a four-cylinder engine. The difference between the engine having a pressure of ½ lb. and those having a pressure of 100 lb. per sq. in. was due merely to differences in the oil-grooves. The engine with a pressure of ½ lb. per sq. in. had a bearing that was all oil-groove, and the one with a pressure of 100 lb. per sq. in. had hardly any oil-groove; so, by determining the common factor and utilizing it, we developed a successful engine. If any one subject should head the list of requirements for practical engine designing, lubrication is that subject.

### EFFECTS OF OVER-OILING

A MEMBER:—What has been the experience with over-lubrication resulting from pressure feed to the wristpins of an engine? In 1917 we had a Buda motor-truck engine which had pressure-feed lubrication to the wristpins and we never were able to obviate smoking cylinders until we closed the tube leading up the connecting-rod. No serious wear on the wristpins resulted, owing to the fact that the rod had on its top a spot-faced hole which allowed the oil to drip down into the top of the rod. That cured the oil-smoking trouble.

CHAIRMAN HORNING:—The tube running up the connecting-rod on the Buda engine was a device imposed by the War Department to assure adequate lubrication at the front in the war. It has been conceded generally that very good results can be obtained by taking oil from inside the crankcase, or taking some oil off the cylinder-wall. With the piston-rings that are in use at present, I think there is very little difficulty due to over-oiling. On the other hand, I agree with the remark made previously that it is better to over-oil and wipe the oil off than to under-oil.

DR. DICKINSON:—Chairman Horning has made the point about the importance of the oiling problem, and there is no question about it. Still, there is a rather amusing phase of the whole discussion to which I would

[9] M.S.A.E.—Assistant chief engineer, Cadillac Motor Car Co., Detroit.

like to call attention. One very successful engine with which I had to deal years ago ran very well with 4 drops of oil per min. to each journal. Another engine required 2½ gal of oil. This is a wide difference between successful engines. Chairman Horning mentioned oil pressures all the way from ½ to 100 lb. per sq. in. There is room for plenty of work on the lubrication problem to see if we can determine some common factor which will serve to correlate such extreme discrepancies between the oiling systems of successful engines.

CHAIRMAN HORNING:—Illustrating the extreme importance of oil, an engine used on certain motorcoaches ran 50,000 miles without having the bearings tightened. Then the company adopted a very effective air-cleaner and the engine ran 100,000 miles. Later, an oil-cleaner was adopted and the mileage increased to 150,000 miles. The company also uses a large-capacity oil-pump. In heavy-motorcoach service many failures have occurred by there not being enough oil delivered to the bearings at the higher speeds, or because some bearing is getting too much and some bearing is not getting enough. Many four-cylinder engines formerly had their camshafts lubricated by the splash system. If with the same oil-pumps oil is fed to the camshafts, a reduced quantity reaches the main bearings. With a four-cylinder engine, having main rods and three camshaft-bearings, there are 10 outlets altogether. Adding the idler-gear bearing to be lubricated makes 11 outlets. In the six-cylinder engine there were 19 outlets. Many of the early failures in engines were due to adding outlets and retaining the same size of oil-pump.

MR. OLDFIELD:—An engine having drip-feed lubrication was very successful in motor-trucks abroad; it ran an average of 125,000 miles between overhauls.

CHAIRMAN HORNING:—That was the Albion engine, used during the war by the British War Department. The Department published a series of observations on these engines after certain periods of service. The Albion engine had a very intricate system of feeding oil one drop at a time or a certain number of drops per minute. The engine had about a 4¼-in. bore and about a 5-in. stroke. The diameter of the connecting-rod bearing was 1½ in., which we would think insufficient in these days, yet the Albion engine had the best record in the British Army. Its lubricating system provided clean oil, having no dilution and no dirt in it. That engine, with that exceedingly small shaft, showed less wear than many shafts having 2¼-in. diameter and ample length.

# THE MISSISSIPPI FLOOD

WHILE the Mississippi drains four-tenths of the entire continental area of the United States, the location of the river is primarily what makes it peculiarly subject to overflow. In the spring a series of storms moving from the southwest over the lower Mississippi drainage area and northeast over the Ohio is an ordinary feature of the climate of the United States. When the precipitation over the Ohio Valley is heavier than usual and the run-off is poured into the already swollen river at Cairo, Ill., the lower river quickly rises above its natural banks.

Of the three great northern divisions of the Mississippi drainage basin, the Ohio, the upper Mississippi and the Missouri, the Ohio is the most important from the viewpoint of flood control. The 200,000 sq. miles that the Ohio River drains is only four-tenths of the area drained by the Missouri, but the rainfall over its basin is heavy and the percentage of this which reaches the river very high. The volume of water poured into the Mississippi River by the Ohio at high water is ordinarily much greater than that received from either of the other two. The Missouri flows down from an area in which rainfall is comparatively limited and the loss by evaporation is large. The banks are soft, but the flood waters are in general confined by high bluffs lying from 1 to 2 miles apart. The Mississippi along its northern reaches is likewise shaped by bluffs and its flow is stabilized by the action of lakes and swamps.

A short distance south of the mouth of the Ohio the Mississippi River has cut its bed through a spur of the Ozark Mountains. From this crooked channel it emerges at Commerce, Mo., and thence makes its way through a vast extent of flat alluvial lands to the Gulf. The bordering hills which mark the former flood channel of the river are about 50 miles apart and nearly 30,000 sq. miles or 19,000,000 acres of land is comprised in the area. The reclamation of this territory from the river by artificial banks or levees dates back to the plan submitted by De la Tour for New Orleans in 1717. Not much had been accomplished when the project was disorganized by the coming of the Civil War and as late as 1880 the river still claimed most of its 50-mile bed during the flood season. Since that time construction has progressed until the levee line measured 1834 miles in 1926. As the system was extended the older levees naturally had to be raised gradually as the height of the river in the narrowing channel increased. The Army engineers reported 1 year ago that the work of bringing the levees to a height sufficient to give assurance against overflow under the worst conditions hitherto experienced was within 10 per cent of completion.

The Chief of Engineers, United States Army, reported in 1926 that the total expenditure in levee protection since the Federal Government became interested in the project in 1882, for an area covering nearly 17,000,000 acres had been $229,000,000, of which $68,000,000 had been expended by the Federal Government. The amount expended by the State of Louisiana from 1866 to 1882 had been a little over $12,000,000.

About 1250 miles of the levee system was reported last year to be up to the grade specified by the Mississippi River Commission. As a comment on the possibilities of the system it is maintained by the Army engineers that a break in a levee built completely according to Commission specifications has never been experienced. Water has run over the tops of such levees this year, however, and it seems clear that they must be much higher if the relief afforded by crevasses this season is not to be counted on regularly.

The preservation of a broad section of this Country, approached in fertility only by the Valley of the Nile, presents a large problem. Each flood recurrence emphasizes the economic importance of this region as an indispensable source of long-staple cotton and other materials. The assistance of the Federal Government is essential.—*Commerce Monthly.*

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# Tires as a Cause of Shimmy

*By* K. L. Herrmann[1]

Annual Meeting Paper *Illustrated with* Drawing, Charts and Photograph

THE author enumerates and describes various interrelated movements of the front end of the car that are commonly known as automobile shimmy. A long list is given of experiments made in an attempt to correct the trouble. These did not produce consistent results but showed that caster angle acts as a considerable influence, while the influence of camber and toe-in seems to be more on tire wear than on shimmy. Lubrication of springs and conditions affecting the free motion of the steering pivots have some influence but the author sees imperfections in the tires as fundamental causes. The nature of the road is also important, particularly of concrete roads having regularly spaced depressions at joints.

Some of the tire imperfections are described and blame is placed on the tire makers for being less thorough in their methods of testing and inspection than are car manufacturers.

A number of shimmy causes are analyzed quantitatively and shown in curves that are arranged to indicate how shimmy is caused when these forces are synchronized. The author expresses the opinion that tire makers should try more diligently to eliminate causes of shimmy within the tires, rather than to suggest that the car manufacturers supply dampening elements to counteract their results.

In the discussion, several representatives of tire manufacturers take the opposite ground, describe some of their tests and experiments and state that tires are manufactured with a very high degree of accuracy considering that they are made from plastic material.

The importance of caster angle was emphasized by manufacturers of axle gages, who said that this angle varies in different car models and is not always kept accurate in manufacture. This item, they said, should be checked by the dealer when the car is delivered and the car owner should have it checked frequently.

The great effect of gyroscopic action as a factor in shimmy was mentioned and demonstrated by a model that showed the sidewise turning of the pivoted road-wheel resulting from gyroscopic action when the axle end is raised or lowered.

SHIMMY has been considered a mysterious phenomenon in a motor-car. A possible reason is that in most experiments unobserved factors had a greater influence than the improvement under test. Study of the action of some car parts producing shimmy tendencies discloses forces so great that the mysteries about shimmy are that it is confined to such narrow limits and such small amplitude; or even that the car stays on the road at all. Fortunately, we have in the car inherent dampening-factors neutralizing these forces, and it is in the reduction of the shimmy forces and increasing the dampening factors that we find our relief.

## What Is Shimmy?

Shimmy consists of various movements of the front end of the car, and the conditions complained of are as follows:

(1) Kick-back or wheel-fight
(2) Tramp and gallop
(3) Incipient shimmy
(4) Violent shimmy

All of these are to a certain extent related. Kick-back begins at low speeds and is started by rough and choppy pavements or badly worn gravel roads. It is usually complained of as beginning at 20 m.p.h. and continuing through the higher speeds, where it seems to become a part of the shimmy movement. The chief objection is that the kick-back in the steering-wheel makes the wheel hard to hold if the steering-gear is not irreversible.

Tramp and gallop come in at higher speeds, approximately 45 m.p.h., and are slow movements, swinging or vertical, noticed at the front fenders.

Incipient shimmy often comes in at about this same speed and is noticed by the driver as a side movement of the radiator cap. It may be accompanied by kick-back or wheel-fight at the steering-gear. This incipient shimmy, however, does not stay in the car but comes in and goes out at intervals.

Violent shimmy is similar to incipient shimmy except that it usually comes in at speeds of 50 to 60 m.p.h. or more and is so violent that the period cannot be driven through. The doors shake open, the radiator cap seems to move from side to side as much as 4 in., and three caps seem to exist instead of one.

## Looking for the Ethiopian

In the study of shimmy the Studebaker research department has conducted a great number of tests. The design features and conditions tried in some of them were as follows:

(1) Vertical pivot at center of wheel
(2) Axles and wheels with off-sets up to 5 1/16 in.
(3) Pivot inclined up to 6 deg.
(4) Spring steering-gear connecting-rod
(5) Spring steering-knuckle tie-rod
(6) Rubber connecting-rod
(7) Rubber tie-rod
(8) All principal makes of steering-gear
(9) Heavily loaded steering-wheel
(10) Various steering-arm and ball locations
(11) Reversed spring-shackling
(12) Changed front-spring-shackle location
(13) Variations in camber, caster and toe-in
(14) Blocking axle to frame
(15) Heavy tubular cross-members
(16) Damper spring working in oil
(17) Heavy radiator-support cross-member
(18) Heavy engine-support cross-member
(19) Various spring-brackets
(20) Four and six-ply tires
(21) Ribbed and non-skid tires
(22) Axle I-beam section unusually rigid
(23) Different types, mountings and locations of snubbers

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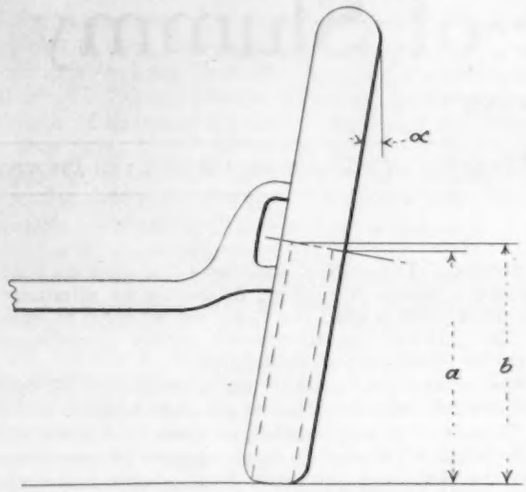


FIG. 1—WHEEL WITH EXCESSIVE CAMBER
As Radius *a* Is Less than Radius *b*, the Corresponding Circumference Is Less and the Wheel Tends To Run in a Circle

(24) Various front-spring deflection-rates from 580 to 250 lb. per in.
(25) Changes in thickness and number of front-spring leaves
(26) Center of gravity changed vertically and longitudinally
(27) Variable passenger-loads
(28) Influence of concrete and ice

These tests did not produce consistent results, but from them the following inferences were drawn:

The chief influence of a caster angle within reason is on turning-effort and the righting of the car after it has been turned. This caster angle should be between 1 and 2 deg., with the top of the axle tipped to the rear. When so set, there will be a slight tendency for the car to straighten itself out after the wheels have been turned to go around a corner, if the steering elements are free, and at the same time it will not be difficult to make a turn. Should the top of the axle tip forward, there will be a tendency for the car to turn, and the driver will have to make a constant effort to keep the car in a straight-ahead course. Should the axle be tipped to the rear more than 2 deg., an extra effort is necessary to make a turn.

### TOE-IN AND CAMBER AFFECT TIRE WEAR

Camber seems to have its chief influence on tire wear. Conditions that cause this wear or shear contribute to wheel wabble and shimmy. Concrete roads at present are built with cambers of 1/10 to 3/8 in. per ft., and the tendency to wear the tires will be least when the plane of the wheel is at right angles to the road contour. Should one wheel be at a greater angle to the road than the other, there will be a constant tendency for this wheel to turn to one side, making the car uncomfortable to drive. Fig. 1 shows the radii of circles representing the difference in the distances two portions of the tread of the tire tend to travel when the camber is excessive. There is a possible influence on shimmy when this camber is not uniformly or correctly set, because it adds to the general instability of the car.

Toe-in is another factor of importance adding to the tire wear and to a certain extent to the instability of the front end. The wheels should be set as nearly straight ahead as is possible under driving conditions, considering all tendencies of the steering-knuckle and bearing to be loose and the direction of the forces when driving. The adjustment usually recommended by tire manufacturers is from straight ahead to 1/8-in. toe-in, which amount seems not to add materially to the tire wear. With improper toe-in and camber there is also a very definite loss of power, proportional to the amount of departure from the correct dimensions.

The most interesting development we have noted in connection with the setting of caster, camber and toe-in has been by the Prather Mfg. Co., of Santa Cruz, Cal. This firm is producing a wheel-aligning machine which, combined with apparatus for setting caster and camber, seems to give a correction for all three elements, as well as an indication of the possible action of the car when on the road. This apparatus consists of two rotating drums on which the front wheels of the car are placed. One drum is pivoted under the wheel until there is no friction or slippage of the tire on the drum. Indicators are attached on this drum to show the amount of correction necessary on the wheel. Also, the drum can be speeded up so that wheel run-out and improper balance can be detected on the stand and the necessary corrections made without inconvenient and indefinite testing of the car on the road.

### TIRES CONTRIBUTE FUNDAMENTAL CAUSE

Other factors influencing shimmy, as developed by our tests, are unsprung weight at front axle; spring lubrication; spring-bushing bind; tightness of shackles; the various periods of the springs and the period of the mass which a spring supports, including the frame, radiator, and engine. The fundamental items, however, in connection with shimmy are in the tire. Some of these are:

(1) Deflection rate of tire
(2) Balance
(3) Air-pressure
(4) Radial run-out
(5) Side run-out
(6) Uniformity of tread thickness
(7) Internal friction in tire

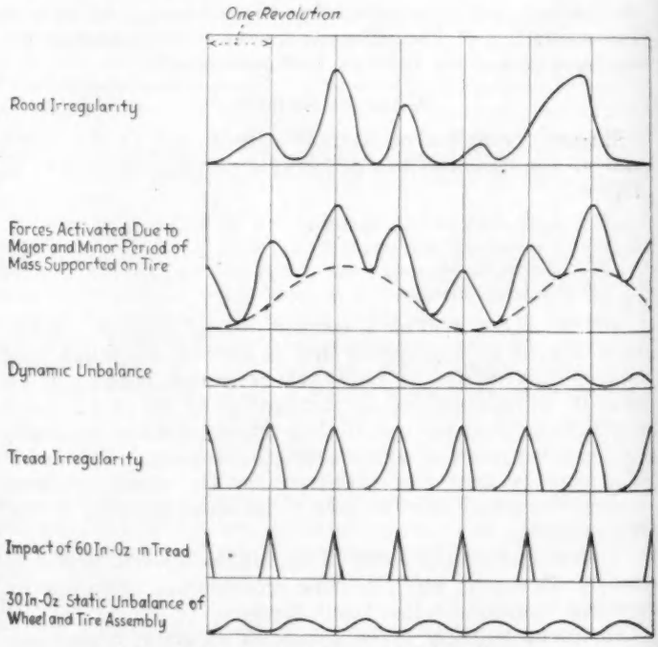
One Revolution
Road Irregularity
Forces Activated Due to Major and Minor Period of Mass Supported on Tire
Dynamic Unbalance
Tread Irregularity
Impact of 60 In-Oz in Tread
30 In-Oz Static Unbalance of Wheel and Tire Assembly

FIG. 2—CURVES OF FORCES CAUSING SHIMMY
These Curves Show the Magnitude and Frequency of Various Forces Tending To Induce Shimmy. Those Depending on Car Design, on Driving Speed and on Recurring Irregularities of the Road Are Shown Synchronized

FIG. 3—LONGITUDINAL SECTION OF TIRE
Overlap of Breaker-Strip Is Indicated at *A* and Overlap of Fabric at *B*. Both of These and the Overlap of the Tread Occurred within About 6 In. in This Case

At present the best balance-tolerance we are able to secure in the casing from the tire manufacturer is 60 in-oz. out-of-balance. The tube is manufactured with approximately 30 in-oz. out-of-balance from the valve. The heavy part of the casing is located and marked with a red square and the valve is located in the casing to counterbalance the heavy side. We have worked with the tire manufacturers approximately 3 years, and they have had considerable difficulty in eventually producing this degree of accuracy. Further improvement in this direction does not seem to be in sight at present. Wheel balance is usually held to 8 in-oz., but no attention has been paid to dynamic balance of wheel or to balance of the rim and the ring where the latter is used. At the top of Fig. 2 is a curve showing the vertical component of the force resulting from this unbalanced condition on the front axle. Dynamic unbalance is represented by the second curve.

## EVALUATION OF CAUSES

Radial run-out of the tire is another factor somewhat beyond our control as the wheel manufacturer requires at least 1/16-in. tolerance and tires vary up to 3/16 in., making a total possible run-out of ¼ in. The force resulting from this run-out is represented by the third curve in Fig. 2, marked Tread Irregularity.

Fig. 3 represents a longitudinal section of a tire showing the overlaps of the fabric *A* and the breaker-strip *B*. Both of these and the overlap of the tread occurred within 6 in. in this particular tire, throwing the major portion of the unbalance to this point. If rotated in free air the wheel could be counterbalanced to offset the centrifugal action at this point in the tread; however, because the tire is in contact with the ground for 10 to 12 in., the effect of this heavy point is removed from the centrifugal forces of the wheel once during each wheel revolution. The result is a reacting force at the opposite side of the wheel, the influence of which is represented in Fig. 2 by the curve of impact. The double curve near the bottom of Fig. 2 represents forces due to major and minor periods of the mass supported on the tires and the springs.

On examining roads which are regarded as shimmy roads, we find that they have regular depressions depending largely on the type and spacing of the joints. Between the joints are other regular depressions which we find measure as much as ¾ in. deep on representative roads, although the principal variations from the level are not more than 7/16 in. The curve of road irregularity, Fig. 2, shows the contour of a road that our car owners and dealers have found induces shimmy when conditions in the car are such as to allow it.

It has been found that correcting some one of these elements will often eliminate shimmy. This is apparently because the change has broken up the synchronism of the various forces described. Attempting to make the same correction which was a remedy in the one car on the next car, shimmy is made considerably worse. Possibly this is because the correction permitted the synchronism of other errors in the second car. It is because of these variables existing in the wheels and tires that the tire manufacturer often supplies us with samples of tires which he has tested thoroughly and found satisfactory, and later, on other cars and under slightly different conditions, these tires do not perform satisfactorily.

## INFLUENCE OF VARIATIONS

Variation of tire inflation seems to cause variation in the cyclic action of the two sides of the car. If one tire is inflated to a lower pressure than the other the shimmy cycle of the car is changed considerably. Another influence is the variable force necessary to flex the various portions in the casing. Possibly the mathematical analysis of some of the forces is beyond our capacity for solution. The curves are diagrammatic and represent only a part of the total forces to be considered.

Combining the curves shown in Fig. 2, we have definite forces of varying intensity, acting at regular intervals and creating nodes such as are found to be very general in our observation on this subject. Fig. 4 is a curve summing up these forces. Our preliminary analysis indicates they are great enough to just about lift the tire off the road. The curve shows one cycle or peak in every seven revolutions in this case, influenced mainly by road irregularities. Similar cycles are produced by the opposite sides of the car working against each other, their periods depending on the uniformity of the pressure in the front tires.

From the above it is evident that the tire manufacturers, who have in addition to supplying us with balloon tires given us shimmy, do not seem to have made sufficiently accurate and definite tests. The initial cost of the tire equipment of a car is approximately half the cost of the engine, and during the life of the car the cost of the tires is considerably more than that of the engine. The motor-car engineer has very definite means for testing and recording phenomena in connection with motor-car performance, including power, torque, economy, frictional losses, wear, engine balance, and vibration, and there is a general distribution of the resulting information. In connection with tires, which we consider an element of fully as great importance as engines,

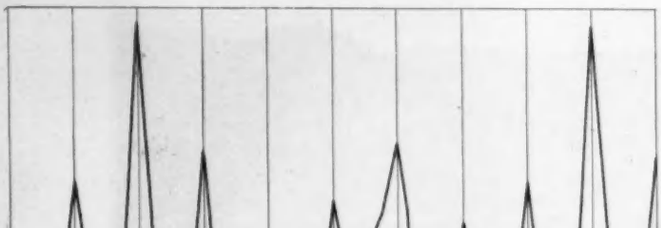


FIG. 4—SUMMATION CURVE OF SHIMMY FORCES
This Curve Is Made by Combining the Curves of Fig. 2. The Highest Points at Each Seventh Revolution Correspond in This Case to Joints in the Concrete Road

there seems to be no definite system or method of testing for tire life as influenced by vulcanizing, fabric, compound and other factors, or for shimmy as influenced by the various elements which create the trouble. The only test-means of which we have knowledge that is used by the tire manufacturer is the motor-car itself, which is entirely inadequate for a reasonable test, just as the test of an engine would be inadequate if conducted only in a motor-car. Also, the tires supplied for test do not represent those supplied for production.

### TIRE PRESSURES

It was only after the most persistent efforts on the part of the Studebaker car-engineers that it was agreed that assembled-tire balance could be maintained to as close an accuracy as 30 in-oz. This was established several years ago and we know of no improvement made since by the tire manufacturer to reduce this unbalance or other defects.

Another consideration is the question of tire pressure. Some years ago, when Studebaker first used balloon tires, the tire companies insisted that these be inflated to not more than 28 lb. It was a long up-hill fight to get this pressure up to 38 lb. and we have not yet succeeded in getting a reasonable variation from this pressure allowed by the tire manufacturer.

Where a tire pressure of 35 lb. is specified, the following variations are likely to occur: A pressure-gage will vary up to 3 lb.; atmospheric temperature variations during the day will change the tire pressure 2 lb.; leakage per day through the valves, permeability of tubes, adsorption of air, and the like often amount to about 1 lb.; and driving the car and warming up the tire changes the tire pressure still another 3 lb. These make possible a total pressure-variation of 10 lb. from causes beyond the owner's control. Yet, where a complaint of shimmy is referred to the tire company, should the tire be inflated 2 lb. under the 35 lb., the explanation is that the tire is underinflated. At the same time it is insisted that we must not inflate this tire to 43 lb. It is obviously impossible for the owner to maintain the 35-lb. pressure specified by the tire manufacturer, but should the pressure drop from the maximum of 35 lb. to 27 lb., the owner driving slowly, the tire manufacturer refuses responsibility if damage is done to the cords due to the low inflation.

[2] M.S.A.E.—Development manager, Goodyear Tire & Rubber Co., Akron, Ohio.

It is getting to the point where owners are beginning to appreciate the inconsistency of our position, and we in turn must refer the problem to the tire manufacturer for solution. It is very evident that, to make a commercial settlement of the problem, the tire manufacturer, together with the wheel manufacturer, will have to revise completely his method of experimentation for determining the influences of the variables in the tire; and improve his control and manufacturing divisions to the extent that the variables outlined may be reduced.

The tire and wheel manufacturers have repeatedly made suggestions as to front-axle design, wheel design and other points as outlined in the list of experiments at the beginning of this paper. It has been established that these various suggestions do not strike the fundamental cause of the trouble, and considerable time has been wasted in trying to overcome with damping and frictional devices fundamental errors that are in the tire.

### THE DISCUSSION

BURGESS DARROW[2]:—The greater part of Mr. Herrmann's paper is devoted to the tire side of the question and I do not agree with him in a number of the conclusions drawn and opinions expressed. R. B. Day, of the Goodyear research department, has read Mr. Herrmann's paper and I should like to quote a few paragraphs from his report, as follows:

> The danger of shimmy or tramping will exist as long as automobiles carry heavy weights in the front wheels freely suspended between upper and lower springs; namely, the chassis springs and the tires. The tendency to shimmy can be altered in magnitude by altering the suspended weights, by altering either or both springs, or by introducing some restraining influence tending to prevent the start of vibration of the suspended weights or to dampen it out if started. No possible change in the tire could remove this condition.
>
> Another objectionable feature that would remain undiminished is the bringing of the center of contact of the tire with the road very near to the point where the king-pin axis intersects the road. This change was made necessary as the most available means of diminishing steering-wheel kick-back caused by unbalanced front-wheel braking. At the same time, this change made it very easy for balloon tires to deflect to either side of the king-pin and thus produce a rapidly reversing steering-wheel whip. This greatly accentuated the tendency to front-wheel wabble. No improvement in tires could possibly do away with this cause of trouble.
>
> I do not see how shimmy in all its forms can be completely avoided while retaining the advantages of the balloon tire without putting some sort of check on the heavy front wheels and brakes, which insist on vibrating between the tires and springs. Putting the pressure in the front tires up higher and higher is merely going back to high-pressure tires and is therefore not a solution of the balloon-tire shimmy problem.
>
> There are three possible objections to the use of horizontal dampers: cost, weight and interference with steering. The many thousands of miles that our drivers have covered with such dampers lead me to dismiss the last of the three objections in toto. The trouble does not exist to any degree that a driver can detect. The other two objections are grim facts; and, in the meantime, so is shimmy.

Mr. Day evidently believes that dampeners will prevent shimmy, and I agree with him whole-heartedly. Tires alone cannot be blamed for shimmy because many cars with balloon tires did not shimmy without front-wheel

brakes, and in many cases the same makes of car with the same sizes of tire now do shimmy with front-wheel brakes. The common use of balloon tires preceded the common use of front-wheel brakes. Tires certainly are not alone to blame.

### Accuracy of Tire Manufacture

Mr. Herrmann's complaint that tires are poorly constructed as to balance must be made without knowledge of all the tire man's problems. A tire is about 7/8 in. thick. Think of a tire which, for a distance of about 15 in. along its tread, happens to be 1/64 in. thicker than the rest and you have a conception of what 20 in-oz. out-of-balance means in tire manufacturing. It means a permissible thickness-variation of less than 2 per cent for only one-seventh of the distance around the tire. Tire companies are offering casing, tube and flap assemblies balanced to within a total error of 30 in-oz. That certainly is not a bad job of balancing, considering that during the process of manufacture we are dealing with a plastic material. Locking the brakes and sliding the tires can get a tire this much out of balance. A repair in the casing or two or three repairs in the tube will do the same thing. Incidentally, the tire manufacturer is more interested in having no heavy spots in a casing than is the automobile manufacturer. A heavy spot in a casing pounds at every revolution and has a tremendous effect on tire life.

If there is a feeling that tire manufacturers have poor facilities for testing, I should like to invite any automobile engineer to come to the Goodyear plant in Akron and see for himself. We have test cars, as Mr. Herrmann says, but that is not half. I will match our equipment and personnel against that of any automobile company in the business, making due allowances for the nature of the product manufactured.

Tire companies are not reluctant to accept advice. I can name dozens of cases where we have accepted advice; but just because one party is the buyer and the other the seller is no reason for the buyer's ideas always taking preference over the seller's engineering experience. I resent any sweeping statement that tire companies are reluctant to take the advice or suggestions of automobile companies and to deal with them fairly.

Balloon-tire pressures are discussed at some length in Mr. Herrmann's paper, reference being made to pressures between 28 and 43 lb., also to variations such as errors in gages, increases in pressure as tire-temperature rises and air leakage, which he says might total 10 lb.

### Recommended Tire-Pressures

Ever since the Rubber Association of America has had a standard load and inflation, in 1925, the following tire sizes and pressures have been shown:

| Section of Tire, In. | Maximum Inflation-Pressure, Lb. per Sq. In. |
|---|---|
| 4.40 | 34 |
| 4.75 | 34 |
| 5.25 | 34 |
| 6.00 | 36 |
| 6.75 | 38 |

I cannot imagine any tire company objecting to these pressures, or even to pressures 2 lb. higher if any automobile company wants them. Such pressures are needed for certain loads; and if an automobile company wants to overinflate up to these pressures for a lighter load and put up with the resulting hard ride, surely no tire company will object.

[3] J. F. Duby Co., Boston.

I will admit that, in the early days, the balloon tire was exploited too freely as a 20 or 25-lb. stunt-tire and cross-sections were too large; but as soon as the tire industry took up the question section sizes were moderated and pressures raised. It is not the tires run at 27 lb. that the tire companies object to and refuse to make good on. For every 27-lb. tire there are dozens of 20 and 16-lb. tires that we have to explain to the customers.

Not much is wrong with our pressure table except that it is so terribly abused. We know the pressure drops from 2½ to 4 lb. per week. Is it unfair to ask a consumer to start out with 34 lb. and re-inflate when the pressure drops to 28 lb., as will happen on the average in 2 weeks? Car instructions advise a user to check the battery every 2 weeks and to oil certain parts of the car about as often.

Tire companies knew all about gages, rise in pressure as temperature increases and valve or tube-leakage when they established the pressure table for balloon tires. Some definite inflation-and-load table was necessary, and the table we selected is very satisfactory when adhered to.

### Large Balloon Tires of Six Plies

I notice that Mr. Herrmann claims shimmy-advantage for the six-ply tire. For a given tire-size and pressure, six plies help shimmy very little. If Mr. Herrmann has in mind a higher pressure for six-ply tires, for example 40 or 45 lb., of course it will help shimmy; but on a light car who wants the kind of a ride that will give?

From the first my company has advocated 6.00 and 6.75-in. tires of six-ply construction, and cars of certain makes have used them for years. The six-ply tire has strength and stamina and will work in the same range of pressures as four-ply tires; but it can, if desired, operate at much higher pressures.

I doubt if many engineers in the industry believe, as Mr. Herrmann does, that shimmy is to be remedied by tire design alone. Plenty of manufacturers are willing to take the problem on their own shoulders as one of car design.

John F. Duby[3]:—Mr. Herrmann's paper would indicate that kick-back or wheel-fight, tramp and gallop, incipient shimmy and violent shimmy are all disagreeable and dangerous conditions; also that we do not know where they come from or what to do about them unless possibly they are caused by the tires, and, in that case, it is up to the tire companies to experiment and find a cure.

I have had a great deal of experience in overcoming shimmy and other steering difficulties. I have worked in connection with hundreds of service managers, have made corrections on thousands of cars and have yet to find one case where the cause could not be located and easily remedied.

### Are We "Passing the Buck?"

It is hard for us to correct a fault if we refuse to admit that the fault exists, and I believe there has been a great deal of shifting of blame regarding this very serious question. The automobile manufacturers blame the conditions on the tires, and the tire manufacturers in turn shift the responsibility back to the automobile manufacturers. So long as this condition exists we cannot look for much improvement. Let us all step in and shoulder our portion of the blame, take our medicine and correct the fault instead of shifting it to someone else.

A good doctor will always diagnose his case carefully

and then prescribe a remedy. Have we done that in regard to shimmy? I think in many cases we have not, yet it is easy to do so if we will make ourselves familiar with the basic cause of shimmy. There are several contributing factors that will aggravate the condition, but without this basic cause I have not found a single case that will develop a shimmy and continue to shimmy for any appreciable time. A car may have practically all of the contributing factors and yet show no indications of shimmy because the basic cause does not exist on that particular car. On the other hand a car may have a very bad case of basic cause and yet, due to all of the contributing factors being eliminated, the car will not shimmy.

As a means of getting down to the base of these conditions, let us consider the facts. First, we had shimmy long before we ever had balloon tires or front-wheel brakes, but since the advent of balloon tires and front-wheel brakes there is no denying that shimmy has been considerably more prevalent. Therefore, it would seem that these two items are not the basic cause but that they aggravate the condition.

Cars of some makes are practically free from shimmying, although they are equipped with balloon tires the same as other cars that are noted for shimmying. We find in several cases where one company is building two models that one model will shimmy very badly while the other is practically free from shimmy. This is not due to the fact that one model is better constructed than the other, because in several cases it is the higher-priced model that does the more shimmying. Some people might say that it is the larger tires on the higher-priced models that cause the shimmying, but we must remember that there are other cars with large tires which never shimmy. Therefore it would seem that there must be something in the mechanical construction of certain cars that makes them more subject to shimmy than cars of other makes. My experience in eliminating shimmy would bear out this theory in every case.

Mr. Herrmann states in his paper that he has stopped the shimmy in several cases, but that when he tried to apply the same remedy on another car it only made the condition worse. This, I think, proves that he has been working on contributing factors. When trying to stop the shimmying on the second car, instead of correcting the one contributing factor that was promoting the shimmy, he changed the conditions so as to bring in another contributing factor. In other words, he did not remove or work on the right contributing factor in that particular case. But had he removed the basic cause the results would be the same on all cars without exception.

I have seen a great many bad cases of shimmying stopped temporarily by tightening the steering-knuckle pivots. This simply causes the pivots to be so stiff that they cannot work up the free motion that is necessary for shimmy. This is similar to taking dope to stop a headache; the cause is not removed. I have seen other cases where the shimmy was temporarily stopped by taking up the play in the steering-gear. The shimmy is stopped temporarily but the cause still exists and it will be only a short time until the shimmy returns.

### Caster Angle Is the Key

Mr. Herrmann stated that in violent shimmy there seem to be three radiator caps where there is normally only one. If you will keep your axles at the proper degree of tilt you are almost sure, in this prohibition Country of ours, to see not more than one radiator cap on the front of your car. In other words, while there are several contributing factors, the basic cause of what we call a real shimmying condition is excessive caster-effect. This caster effect which causes shimmy is a condition that is desirable to make steering easy, but we must remember that even a small amount of caster effect will cause shimmy providing all of the other conditions are favorable to shimmying. Therefore it is important to adjust all of the contributing factors to somewhere near the ideal conditions so that we may have a little caster-effect to facilitate steering.

On one car which is practically free from shimmy the steering-gear arm is 9 in. long and its girth is 3⅜ in. at the center. Another car that is noted for shimmying has a steering-gear arm 12 in. long, and the girth of this arm at the center is only 2⅝ in. On this car the slightest inaccuracy in caster action will cause shimmy because the steering-gear arm is not stiff enough to hold the wheels in their course.

In treating shimmy I have always separated what we call genuine shimmy from tramp or gallop. Back in 1916, long before we heard of balloon tires, I was agent for a car which, although good in many ways, was exceptionally hard riding. A friend bought one of these cars and was proud of the fact that on a certain road he could drive this car at a certain speed and make it gallop. In fact, he used to take his friends to ride over this particular road so that he could make his car perform for them. The road, although smooth, was slightly wavy; and the galloping of which he was so proud was entirely due to his driving at one certain speed so that the waves on the road, which were regular, would be encountered at the same period as that of the recoil of the springs. This condition should not be confused with shimmy, as it is an entirely separate condition, but it certainly is not caused by balloon tires.

Another condition that is sometimes confused with shimmy is brought about by an unbalanced wheel. There is no doubt that a badly balanced wheel will cause a racking motion at high speeds. If both front wheels are badly out of balance this condition is greatly aggravated. If the two heavy parts of the wheel hit the road at the same time we get a condition similar to galloping, but if the two heavy points come in contact with the road at points approximately 180 deg. apart we get a condition very similar to shimmy. If we add excessive caster-effect the condition is made worse.

In all of my experience I have not yet run across any case where the tire itself was badly enough out of balance to cause shimmy or gallop unless the heavy part of the tire happened to be placed at or near the heavy side of the wheel. If the wheels and rims are properly balanced, with a weight on the opposite side to counterbalance the valve and valve-stem, I think we do not need to worry about the balance of the casing on any standard make of tire today.

### Influence of Steering Connections

Another condition that is sometimes confused with shimmy is what we call whip, also known as a fighting steering-wheel. With the conventional type of front-axle assembly the front spring acts as a radius-rod approximately 20 in. long. When the car encounters bumps that cause the springs to flex, the axle swings on this radius while the steering-gear connecting-rod swings from the rear ball-joint on a radius that may be 30 in. If the front end of the rod is forced to follow the arc described by the axle instead of traveling in an arc from its own rear mounting, the result will be a constant

tugging and yanking on the steering-gear while traveling on rough roads. If the front ball-joint is kept close to the center line of the two arcs, this tugging will be at its minimum. With a 1½-in. travel in each direction the total change in length will be only 1/10 in., but if the front end of the steering-gear connecting-rod is set 1½ in. above this line a 1½-in. flexing of the spring in each direction will cause a shortening of the distance between the front spring-bolt and the rear ball-joint of approximately 4/10 in., and this shortening effect must be compensated for either by the road wheel turning slightly to one side, the steering-wheel turning in the hands of the operator, a deflection of some of the steering connections, or possibly by a combination of all three.

You will readily see that this sudden jerking of the wheels to one side when we encounter a bump on the road can easily be the means of starting the car to shimmy, but it cannot possibly cause continued shimmying unless there is excessive caster-effect.

I do not think that we should say the tests made by the tire manufacturers are of the crudest possible type. As a matter of fact the tire manufacturers have come to the rescue several times by correcting conditions that, in my estimation, could and should have been corrected in the car itself. There has been a crying demand for easier-riding cars for the last 20 years. I believe that the riding-qualities could and should have been put into the springs, but, in spite of the announcements of the various manufacturers from year to year that they had lengthened their springs and thereby made the car ride easier, we still have springs of approximately the same length as they were in 1910. The tire companies, seeing this demand for easier-riding cars, produced the balloon tires and there is no question that they have given us easier riding.

### Accurate Settings Needed

Balloon tires and front-wheel brakes have, however, made it necessary to maintain a greater degree of accuracy in the setting of the axle, and to do so we must have the proper means of ascertaining these settings. I have had men tell me that they changed the degree of axle tilt, thereby changing the caster action, and that it made the car shimmy worse. When asked what means they had of testing the amount of tilt, they said they guessed at it. If we are to stop the shimmy we must first determine the exact amount of tilt that exists and then insert shims to bring it to the correct angle.

We must remember that we can sometimes make a car shimmy worse by slightly reducing the amount of caster action, for between 3 and 4 deg. seems to be the worst point. Therefore, if an axle has 5 deg. of tilt and wedges are put in to bring it down to 3½ deg., it is very likely to make the car shimmy worse. Another condition that must be avoided is the possibility of insufficient tilt or of slanting the axle and pivots in the wrong direction, as this would cause what is called buckling under, which is even more dangerous than shimmy.

I have mentioned only a few of the contributing factors but it can be seen that the importance of having an accurate means of testing for these conditions cannot be overestimated.

### Experiment Results Not Universally Applicable

W. R. Griswold[4]:—The author of this paper has been prudent enough to say that the results of the 28 reported experiments which his organization conducted were results as they determined them and does not suggest that they be held as universal conclusions.

In the matter of shimmy each car presents a totally different problem from a car of any other design, and often slight differences occur in production, making some cars slightly different from others of the same model. Therefore conclusions reached by experiment with one model generally have no bearing on the behavior of another model when a similar set of experiments is made. The phenomenon of shimmy, when reduced to its fundamental aspects, evidently follows the definite laws of the general class of dynamic problems in which it falls; namely, that of a vibrating system. Four factors enter into the shimmy problem. They are the forces that excite or start the vibration, the restoring forces, inertia and damping forces. Unbalanced wheels, gyroscopic effects, road inequalities and faulty steering-geometry cause exciting forces. Elastic elements, including the frame and the springs and all parts of the steering mechanism that are subject to deflection, must be considered; also the inertia of all parts entering into vibration. Damping comes mainly from the friction in the mechanism. The phenomenon of shimmy may be expected under any condition where there is not sufficient damping action to overcome the exciting force.

### Gyroscopic Effects

A peculiar thing about the shimmy problem is that the gyroscopic forces are not primary exciting forces but are produced from them as secondary exciting forces. One of these primary exciting forces is brought into play because of poor steering-geometry, and the effect of this can be exceedingly large when the geometry is much in error. One way to overcome this is by the use of front-shackled springs in combination with a steering linkage arranged so that pivoting of the knuckle due to any vertical compression of the front spring will be the practical minimum. Simply shackling the front end of the front spring is not itself sufficient; it is also necessary to have the geometry right. By eliminating or greatly reducing this primary exciting force, the occurrence of shimmy is considerably lessened.

The forces due to unbalance of wheels and tires are primary exciting forces of great importance, and too much emphasis cannot be placed on the need for having balanced tires. The forces due to unbalance of the tires and wheels are of a harmonic nature just like the gyroscopic and the elastic forces. The primary exciting forces may be sufficient to overcome the friction in the steering mechanism and thereby give rise to the growth of the gyroscopic force which may start a wheel wabble accompanied by some evidence of radiator shimmy, or start a tramping or radiator shimmy accompanied or unaccompanied by wheel wabble. The gyroscopic forces are generally much greater than the primary exciting force of unbalance. This is especially true if the unbalance is limited to small amounts. At high speed shimmy and tramp are essentially the result of the timed relation or resonance between the gyroscopic or secondary exciting forces and the elastic forces. However, even the elimination of all wheel unbalance and all steering-geometry error is no guarantee that shimmy will be eliminated, although the intensity of the shimmy will be greatly reduced. It is still necessary to have more or less effective damping, and this is obtained essentially in the front springs.

Such things as the flexibility of springs or the flexibility or stiffness of steering mechanism have only the influence of locating the speed at which shimmy occurs.

[4] M.S.A.E.—Engineer in charge of analysis of design, Packard Motor Car Co., Detroit.

Frame stiffness, especially of the side members and the cross-bracing, is extremely important in establishing the location of the shimmy in the speed range and controlling the amplitude of the shimmy if it does occur within the driving range.

### The Remedies

With the conventional front-end constructions, the most practical ways of overcoming shimmy may be enumerated briefly as follows:

(1) Balance wheels and tires to close limits
(2) Design the steering system for minimum error of steering geometry
(3) Reduce the flywheel effect of front-wheel assemblies to the minimum
(4) Use stiff frame-construction, especially as to torsional stiffness
(5) Take advantage of all possible damping action, such as interleaf friction of the front springs
(6) Make the stiffness of the steering linkage as great as possible, consistent with weight and other restrictions

B. J. Lemon[5]:—The great art in writing is not so much to exhibit one's own insight and learning as to influence those whom one is aiming to influence, by suggestion, stimulation or irritation. This is particularly true in the field of automotive engineering. The engineer has a certain professional familiarity with some special realm of knowledge. When he wanders beyond this field, he often assumes a certain sort of outlawry to stimulate the necessary reaction in the minds of experts in the transgressed fields. The author of this paper has unquestionably produced a reaction from the tire industry and, therefore, to that extent, has written a successful paper.

It is regrettable that more space could not have been assigned to toe-in, pitch and caster of motor-car front-wheels and axles. The industry as a whole knows far too little of the mechanics of alignment as affecting steering, tire wear, shimmy, and general stability of car operation. Alignment-instrument makers say that shimmy can be largely eliminated by proper front-wheel alignment and axle tilt. Rumors, admittedly unsubstantiated but nevertheless decidedly disconcerting, are widespread that the amounts of toe-in, pitch and caster used on new cars going from factories do not accord with published specifications. These rumors should be verified or refuted by a frank and understandable discussion of the whole question, including a study of the methods and instruments for properly measuring alignment and making corrections when misalignment is found. We must meet the problem by holding alignment specifications within such limits as will be practical yet will not stifle future development. To the tire maker, 1/16-in. toe-in and ¾-deg. pitch appear to be ideal specifications.

### Tire Balancing

One impression conveyed by the paper is that the tire maker not only has failed to study his tire from the standpoint of shimmy or balance but that he is woefully backward generally in respect to methods and machinery for tire testing. Three years ago this month our company began to balance tires. Special balancing machinery was designed for this work. At first production deliveries were held at 60 in-oz. for the combined casing, tube and flap. Since then we have continuously improved our methods of balancing and studied tire construction to improve tire balance. A few of the construction features investigated were:

(1) Tread design
(2) Angle of cords
(3) Flipper construction
(4) Bead construction
(5) Breaker width and fabrics
(6) Thickness of rubber between plies and of sidewall rubber
(7) Spacing of splices in carcass, breaker, fabric and tread

As a result we are now holding the balance-error of our product within 30 in-oz., a reduction of one-half, and we have not given up hope of improving this.

An especially stiff tire-sidewall decidedly advances the minimum speed at which shimmy will be experienced, but such construction is used at the expense of riding-quality. In this category falls the six-ply tire.

The author of the paper admits a lack of acquaintance with other than road-test methods of proving tires. The tire maker offers no apologies for his road-testing program, whether he is testing for shimmy or for tire endurance. This test is unquestionably the most acceptable and practical test developed up to date for both automobile and tire. If it is not, why is the industry building special proving grounds, the purpose of which is to speed-up road tests?

### Tests Applied to Tires

But the present methods of testing tires are by no means limited to road tests. An incomplete list of other tests which we apply to finished tires is:

(1) Measurement of at least 10 different dimensions
(2) Cure tests on nine major parts
(3) Physical tests, about 35, such as tensile, flexing and elongation
(4) Bruise tests
(5) Nail-puncture tests
(6) Hydraulic strength-tests
(7) Tread-abrasion tests
(8) Deflection tests under load
(9) Profile, tread-impression and tread-load-distribution tests
(10) Power-consumption tests
(11) Noise tests
(12) Run-out tests
(13) Balance tests
(14) Throw-off tests
(15) Test-wheel tests
(16) Oscillating tests
(17) Steering-torque and traction tests
(18) Complete examination of returned tires

Radial run-out of tires is cited as a fundamental cause of shimmy. This is taken to mean the radial variation in the section of a tire measured inflated and at no load. Such variation probably is due to the splicing of the carcass and breaker fabrics and tread and to the tube splice. The inflated casing may be slightly smaller at such points. Three-sixteenths of an inch is given as a possible variation on, we assume, an unloaded tire.

In our opinion this method of measuring run-out leads to erroneous conclusions if its effect on shimmy is under consideration. The effect of run-out should be measured under load and is so measured in our testing program. As an example, we have measured for deflection under a 1300-lb. load, a 6.20-in. casing that was 60 in-oz. off balance. The section measurements were taken under load at various points, including the points of greatest and least radial run-out. With a total deflection of 1.312 in. the greatest variation in deflection was 0.025 in. This

[5] M.S.A.E.—Automotive contact engineer, United States Rubber Co., Detroit.

variation is so small that it is believed to have no appreciable effect on shimmy, or at least so slight an effect as to be practically negligible when compared with the effect produced by road irregularity, admitted to be as great as ¾ in. on an average road.

The above deflection covers static conditions. At 45 m.p.h. the greatest increase in radial run-out is 0.12 in. on a tire with no load. We have made no measurements of radial run-out for loaded tires at 45 m.p.h.

Lateral run-out we assume is any irregularity of carcass or tread that will produce lateral tire-wabble. Lateral run-out probably slightly increases radial run-out. One-eighth of an inch is an abnormal maximum and 1/16 in. is an average static lateral run-out for a well-constructed automobile-tire mounted on true rims. This may be increased 0.05 in. at 45 m.p.h. We have not been able to measure any increase or decrease of shimmy due to lateral run-out of tires.

The average thickness-variation of the combined tread and carcass of our 5-in. tires is 1/32 in., measured under no load. This variation may be slightly greater for larger tires. The effect of this variation is covered under radial run-out.

The day-to-day variation in shimmy characteristics of a car or tire is one of the elusive things about shimmy. Engineers are acquainted with such variation in the same car with the same set of tires despite every possible effort to hold all variables constant.

They know that if weights, wheels, tires, springs, steering mechanism, and amount of unbalance are all held constant; if the road, speed, weather conditions, temperature, moisture, and wind are maintained as nearly identical as possible, even then shimmy is a day-to-day variable. The tire is a flexible envelope, enclosing a fluid gas. We do our best with the material we have to use but are limited thereby.

As to the imputation that shimmy is the unfathered offspring of the tire, we can point to the shimmyless two-wheel-brake car as satisfactory evidence that the responsibility for the conception of shimmy must be borne equally by the front-wheel brake.

Tires supplied by us for test represent standard production unless special sample construction is requested for experimental purposes where test for shimmy or some other experiment is involved.

We have not found inflation-measurement gages to vary more than 1½ lb. within the range of balloon-tire inflation-pressure. Our experience is that the maximum variation in inflation pressure, including gage variation, might be 6 lb. rather than 10 lb. above or below the recommended figure under average operating conditions.

We have counseled against using more than 36-lb. air-pressure in four-ply front tires but have never refused to agree to 40 lb. when the question of shimmy was involved. Our records show that 3 days after we received a request to approve more than 36-lb. pressure we agreed in writing to stand back of our four-ply tires up to 40 lb. of air. This figure of 40 lb. is exclusive of any variations due to gages or to other causes.

The reasons for desiring to limit the maximum inflation were to retain the cushioning properties of the balloon tire and at the same time assure against high-speed bruising that may result from abnormally high inflations in four-ply tires. Thirty-six-pounds inflation in front tires on all present-day cars means less than 15 per cent tire-deflection, which is approximately the deflection of the high-pressure tire previously used. Since shimmy was a rare occurrence with high-pressure tires on two-wheel-brake cars, the fact that we get shimmy on four-wheel-brake cars at 36-lb. inflation is strong evidence that front-wheel brakes are a major factor.

Our company has carefully avoided suggesting cure-alls or special remedies for shimmy; but because we have not dissipated our energy on such flank movements it does not follow that we have neglected to study our own product.

**The great art in writing is not so much to exhibit one's own insight and learning as to influence those whom one is aiming to influence by suggestion, stimulation or irritation. This is particularly true in the field of automotive engineering. The engineer has a certain professional familiarity with some special realm of knowledge. When he wanders beyond this field, he often assumes a certain sort of outlawry to stimulate the necessary reactions in the minds of experts in the transgressed fields. The author of this paper has unquestionably produced a reaction from the tire industry and therefore, to that extent, has written a successful paper.**

Important accelerating forces generated by the wheel have not been prominently mentioned in the catalog of shimmy causes outlined in the paper under discussion. One of these forces, the gyroscopic couple, appears to lift one wheel and depress the other if the wheels tend to wabble. Gyroscopic forces are accentuated by front-wheel brakes because of the greater weight and because the center of weight of the front-wheel unit is brought closer to the steering-knuckle pivot. Such accelerating forces were discussed[6] in detail at the 1925 Annual Meeting of the Society.

A universal solution of the shimmy problem has not made its appearance except perhaps through the adoption of higher air-pressures or stiffer tires. The trend to tires of lower standing-height has helped because such tires are stiffer and the center of mass of the car is lowered. The reduction of weight in front-wheel brakes has also aided. Alterations in steering hook-up, in snubber attachment, in wheel alignment and in caster all have a bearing. We are gradually getting a better diffusion of knowledge, a closer exchange of facts, which is a great help in the solution of any problem.

H. W. DAMMANN[7]:—We have found from actual experience and tests that many new cars have left the factory with their wheels and axles not set according to specifications. Some manufacturers check the wheels for toe-in but few of them pay any attention to the pitch, which we find is just as important if not more important than the toe-in, especially with balloon tires and four-wheel brakes. We have found also that very few manufacturers are in a position to measure caster or tilt in the front axle. We found in a number of cases that the caster in the front axle was several degrees wrong, and the manufacturer did not know it and would not acknowledge it after he was shown. We know of cases where we corrected this for the manufacturer and overcame the trouble, yet these same manufacturers do not care to recommend equipment to their dealers that would enable them to overcome their difficulty. Instead of educating their dealers they try to make them believe

[6] See THE JOURNAL, June 1925, p. 624.
[7] Bear Mfg. Co., Rock Island, Ill.

their cars will not shimmy, and consequently their dealers are kept in the dark and are not in position to handle this matter of service today.

### What the Dealer Should Do

When dealers receive new cars they tune up the engines, adjust the brakes and clutches and see that everything else is all right, but pay very little attention to the wheels and axles so long as they run and stay in the road.

When new cars are put out with the front end out of balance it is only a matter of a short time before they begin to come back with complaints of steering, tires worn out or shimmy. The dealers try to overcome this trouble by checking the wheels for toe-in and pitch, but if this does not remedy the difficulty most of them are through. The easiest way out is for them to blame the tires, as they do not want to shoulder any responsibility. The dealer who is not in position to check the caster of the axle as well as its pitch and toe-in cannot expect to handle this service successfully.

Most axles are tilted backward, that is, the top of the axle leans toward the rear of the car. Each manufacturer has his own specifications, and they vary greatly in different makes of car. We find that with front-wheel brakes and balloon tires it is easy to knock the axle out of alignment, and unless the dealer knows how and has the necessary equipment it is impossible to get it correct. We also find that one front brake grabbing more than the other sometimes causes such violent stress as to twist the axle and thus upset the balance between pitch, toe-in and caster. These are simple to adjust providing the mechanic knows exactly what is at fault and how to correct it.

The remedy is simply a matter of education and equipment. The dealer must be able to maintain the specifications as furnished by the manufacturer. You will never be able to get entirely away from readjusting because some cars will come back with their front ends knocked out of balance through some cause. However, if the car manufacturer will require his dealer or distributor to check over the pitch and toe-in of the wheels and the caster of the axle on every new car before he delivers it to the customer, most of this trouble will be eliminated.

F. F. Chandler[8]:—When front axles are damped with snubbers I have observed that these act in a way to distort the geometry. When spring deflection is caused by impact, the snubber acts with it. When the spring deflects due to a rebound, snubber action resists it. The front-spring deflections can in this way be actually quite different from those that are expected, and this sometimes causes difficulty.

Mr. Duby:—I might answer that by saying that we have found that it is better, in cases where shock absorbers are installed, to have the steering-gear connecting-rod slightly below the center line. That allows it to work equally across the center line in ordinary driving, as the snubbers prevent the spring from going so far in one direction as they do in the other.

J. S. Bushey[9]:—Three years ago we were called upon to change from high-pressure to balloon tires. After making the change-over of about 100 sets, we were confronted with the shimmy difficulty. In addition, on account of road conditions in California, 95 per cent of all mileage is covered on reinforced-concrete pavements and an almost perfectly flat surface. On the surface, where the tire comes in contact with it, the pavement is rough and extremely hard.

### Some Field Experience

When we placed balloon tires on automobiles that were built for high-pressure tires we had what is termed side-wear. Whether it is on the outside or the inside, this is not a natural condition. It has a great deal to do with wandering, romping and these other things. We took up this difficulty with men who seemed likely to have had the same experience but were not able to obtain a definite solution of the problem. The suggestions we received from the engineers did not cure the trouble. So it was necessary for us to either close our doors and leave our territory or else go ahead and whip the problem. I do not want to intimate that this was the problem of the car manufacturer, the wheel manufacturer or the tire manufacturer; it was the problem of the man who put the wheels on and lined them up.

The question of tire contact on the road has not been touched upon. Assume that the wheels are balanced and carefully mounted on an axle that is strong enough to carry the load. It is necessary to have pitch or camber in the axle. To get a very close adjustment, the amount should be determined at the time the car is assembled, using a gage that is graduated closely enough to find any error. The axle should then be adjusted with the weight of the car on the tires. Taking 150 cars, 25 with straight wheels and other lots of 25 with camber from 0 up to about 1 in., and running them 5000 miles, we found with few exceptions that we could get better tire-mileage and avoid shimmying and friction by adjusting with ½-in. camber in each wheel and ⅛-in. toe-in. Caster is adjusted to make the car handle as it should. The same condition, with allowance for larger or smaller wheels, applies not only to the passenger-car but to the motorcoach and the motor-truck. We find that solid tires will sometimes, the same as pneumatic tires, wear at one side.

Mr. Lemon:—Do you find any objection on the part of the car manufacturers to bending of the axle?

Mr. Bushey:—We discussed the matter of axle bending with the axle manufacturers about 3 years ago and they heartily endorsed our method. In fact, we are adjusting the axle in the same manner as are the manufacturers whenever they find it necessary after heat-treatment. We always bend the axle cold, as approved by the axle manufacturers. There is no danger of bending the axle enough to weaken it. It will be bent only 1/16 in. at the most, and, as it is done by men who know the work, there is no difficulty or possibility of weakening the axle.

Mr. Lemon:—The Tire & Rim Association has been trying for about 3 years to agree upon the proper toe-in and pitch of wheels as they affect tire wear. I believe they have recently submitted their ideas to the chief engineers of the manufacturers of cars, trucks and motor-coaches, suggesting 1/16-in. toe-in and ¾-deg. pitch.

A. S. Van Halteren[10]:—We investigated to see how much our wheels were affecting this problem. We find that the wooden wheels will run within 25 in-oz. and will average about 15 in-oz. out of balance. Steel wheels will run within about 10 in-oz.

H. A. Huebotter[11]:—Gyroscopic action of the wheels and axle was mentioned by Mr. Griswold as a contribut-

[8] M.S.A.E.—Chief engineer, Ross Gear & Tool Co., Lafayette, Ind.
[9] M.S.A.E.—Owner and manager, J. S. Bushey Co., Los Angeles.
[10] M.S.A.E.—Chief engineer, Motor Wheel Corporation, Lansing, Mich.
[11] M.S.A.E.—Associate professor of gas engineering, Purdue University, West Lafayette, Ind.

ing cause of shimmy. I have here a model that represents a front wheel mounted on an axle from which you can see the effect of this action well enough to recognize that a tramping axle is sufficient to cause shimmy. If you are interested in the mathematics involved I will be glad to present it as a written contribution[12] to THE JOURNAL.

Methods to damp out the tramping can be devised. Anything to break up the period of the axle or to increase the frequency beyond the driving range should be effective.

[12] See THE JOURNAL, April, 1927, p. 423.

MR. HERRMANN:—Our friends from Los Angeles have contributed various definitions that are very vital, and Mr. Huebotter has presented a vivid demonstration of gyroscopic phenomena with which we may not be familiar. In the discussion of this subject I have been attacked from an unexpected point. Also the gage makers gave me a surprise. Their business has grown since the advent of balloon tires. Gages are good things but we must be on guard against flimsy gages that are easily bent so as to be inaccurate. The tire men say the tires are all right and recommend some dampening apparatus, just as I expected them to do.

# MONEY STOCK IN THE UNITED STATES

THE stock of money in this Country today is twice what it was before the World War. On May 1, 1927, the inventory of the stock of money in the United States showed $8,600,000,000. Of the increase of almost $4,900,000,000 since 1913, $2,700,000,000 is represented by gold and $2,100,000,000 is supplied by Federal Reserve notes, of which there was none before the establishment of the Federal Reserve System.

The figure for United States notes, commonly known as greenbacks, has not changed for nearly 50 years. No large change has occurred in the amount of national bank notes in recent times. This amount has been largely governed by the amount of United States Government bonds outstanding which bear the circulation privilege. Of these there are now only about $675,000,000. Almost $75,000,000 of these consist of two Panama Canal issues callable at any time.

Before the passage of the McFadden Act. the Treasury announced that the privilege of emitting national bank notes would probably be withdrawn by degrees after national banks were given compensating advantages of operation by law. Certain national banks have been reluctant to give up this privilege because it carries with it the possibility of a small profit as well as sentimental and advertising value. The method for accomplishing the retirement of national bank notes is through the retirement of the bonds bearing the circulation privilege which underlie them.

With the Panamas retired, there will remain only the Consols of 1930 bearing the circulation privilege. If these are retired at maturity in 1930 without the emission of other bonds bearing the circulation privilege, the national bank note should completely disappear from our money stock. The Treasury has stated that this would simplify our currency and is in line with the policy inherent in the Federal Reserve Act.

MONEY STOCK IN THE UNITED STATES ON MAY 1, 1927

| | |
|---|---|
| Gold Coin and Bullion | $4,600,000,000 |
| Silver Dollars | 500,000,000 |
| Subsidiary Silver | 300,000,000 |
| United States Notes | 400,000,000 |
| Federal Reserve Bank Notes | 5,000,000 |
| Federal Reserve Notes | 2,100,000,000 |
| National Bank Notes | 700,000,000 |
| | $8,600,000,000 |

The Reserve banks were authorized to purchase a certain amount of circulating bonds each year, against which they were to be allowed to issue Federal Reserve bank notes. During the war the Reserve banks bought a relatively small amount of these bonds and then ceased purchasing.

### PAPER MONEY NOT INCLUDED IN STOCK

In inventorying the stock of money in the Country, gold and silver certificates and Treasury notes of 1890 are not listed by the Treasury. This is because this paper money is merely the circulating form for gold and silver. A silver dollar must be impounded in the Treasury against each dollar of silver certificates or Treasury notes emitted, and a dollar's worth of gold must be impounded against each dollar of gold certificates emitted.

Silver dollars in the money stock are a few million less than before the World War. The contract between the United States Government and Great Britain for the purpose of bolstering Indian exchanges caused the silver-dollar stock to go down by more than 250,000,000 during 1918 and 1919, when silver dollars were melted and sold to Great Britain. But an equivalent amount of silver has subsequently been purchased for recoinage and the volume of silver dollars now is substantially the same as before the World War.

On the whole, national bank notes, United States notes, Federal Reserve bank notes, gold and silver certificates, and Treasury notes represent inelastic elements of our currency. The major fluctuations in our money stock during the last few years have been predicated on the change in the stock of gold and Federal Reserve notes. When all of the other types of money are lumped into "All other, miscellaneous," a surprising constancy in this aggregate is displayed over the years in question.

Gold is, of course, the basis of our system of values and its supply is conditioned on international exchange relations. While artificial devices may cause it to flow in or out, the gold supply is assumed to be controlled by economic law and is not generally regarded as elastic.

The really elastic element of our currency is the Federal Reserve note. To provide for elasticity, theretofore lacking in our money system, was one of the main purposes of the Federal Reserve Act. The machinery of the emission of Federal Reserve notes, with their security of commercial paper and gold, and how their volume should respond to the money needs of the Country are the subjects of frequent rehearsal in financial circles. The Federal Reserve note first appeared in our currency in 1914, the year the system was put in operation. As of June 30, 1915, there were $84,000,000 outstanding; by June 30, 1920, $3,400,000,000. There has been a general decline since 1920 in the number of Federal Reserve notes outstanding. On June 30, 1925, they stood at $1,942,000,000 but had increased somewhat by May 1, 1927.—*Commerce Monthly.*

# Light-Alloy Pistons

*By* G. D. Welty[1]

Buffalo Section Paper *Illustrated with* Drawing

PISTON troubles are not confined to pistons of any one material, and they can be overcome by proper design with either alumiunm or cast iron. The light weight and high heat-conductivity of aluminum alloy are advantages, and difficulties due to its high coefficient of expansion from heat can be overcome by proper design.

The development of permanent-mold pistons with split skirts and with steel or Invar struts to control expansion is sketched. Provision for lubrication of the cylinder-wall immediately after starting the engine is described; and working clearances for piston-rings are recommended. An outline of the status of magnesium-alloy pistons in this Country at the present time is given.

Early experiences with aluminum pistons are related in the discussion, and the problems of aluminum pistons for Diesel engines are stated. Practice as to piston-ring width is reported, and recommendations are made as to methods of securing the piston-ring in place. The suggested use of aluminum for piston-rings is not received favorably.

THERE is an engineer of a large automobile company who has always taken delight in "heckling" me whenever opportunity offered. He said not long ago, "I wish you would tell me what is really wrong with aluminum pistons." I said: "There are only three things that are really wrong with them; they stick, and they slap and they pump oil. Now you claim to be a great expert in design and an advocate of cast-iron pistons even though you do not use them; suppose you tell me what is the matter with them." He said, "Cast-iron pistons also have just three troubles; they stick, and they slap and they pump oil."

We must go deeper than sticking, slapping or oil-pumping to justify or to condemn either cast iron or aluminum as a piston material. Sticking, or "freezing" as it is sometimes called, is more serious with cast iron than with aluminum. Slapping is more frequently encountered with aluminum. Oil-pumping, under certain conditions, is common to both. These are but complaints or annoyances due to improper construction or adjustment. They are more or less serious, but they are successfully dealt with in either type of piston. Such difficulties, while very influential in creating prejudices, do not represent the fundamental considerations and they will not, in the long run, settle the question. That will be settled according to which material most economically meets the requirements.

What are the requirements of a piston? A piston should be made of material having excellent machining properties, inasmuch as it is finished all over. It must provide a good seat for the rings and a perfect bearing for the piston-pin. It must act in the engine as both a gas-tight plunger and a cross-head. It should therefore be strong and able to maintain its shape at high temperatures and should have good bearing and wearing properties. The piston must also provide a path for the transfer to the cooling water of a considerable part of the heat generated in the combustion-chamber. The material possessing the best heat conductivity will best meet this requirement. Since the piston is a reciprocating part, its weight should be kept as low as is consistent with the requirements of strength and conductivity. Lastly, although subjected to greater extremes of temperature than any other moving part of the engine with the exception of valves, the piston should perform all of its duties with the minimum of vibration, noise and other difficulties attributed to faulty pistons.

[1] M.S.A.E.—Engineer, Aluminum Co. of America, Cleveland.

## Light Weight and High Conductivity

How does aluminum, by which is meant the light aluminum piston alloys, meet these requirements? These alloys possess two characteristics which make them most desirable for use as piston material. They are light and they have high thermal-conductivity. The advantages of a light-weight piston for reducing vibration and securing smoothness of operation in four-cylinder engines are well known. In six-cylinder engines the increased smoothness due to light-weight pistons is also very apparent, chiefly at the higher speeds. In eight-cylinder engines our experience is more limited, but the effect is apparently to either eliminate entirely, or to reduce very much, vibration periods which appear at certain definite speeds. This results in better bearing conditions and a tighter engine.

The advantages of high heat-conductivity are familiar to the engineer but are not so well known to the public. Chief among them are the possible employment of higher compression-ratios with less carbon deposited on the piston-head, resulting in less preignition, cleaner and cooler oil and cooler bearings. In addition to these things, the aluminum alloy possesses excellent machining properties, is hard and strong, and its wearing qualities are superior if it is properly lubricated.

Aluminum has one property that is unfortunate for its use as piston material. That is a coefficient of thermal expansion more than twice that of cast iron. This property has been the greatest single obstacle in the way of the general adoption of aluminum pistons. If a plain cylindrical aluminum piston, similar to the conventional cast-iron piston, is fitted into the cylinder with the usual clearance allowed for cast iron it will expand so much that it will stick and score at high speeds or under heavy loads. If it is fitted with sufficient clearance to prevent this it will be so loose in the cylinder that it will rattle back and forth when the engine is started cold, producing the noise known as piston slap.

To overcome this difficulty is the dream of every piston inventor. The vast majority of all of the 1000 light-alloy-piston patents deal with some scheme for compen-

sating for the greater expansion of aluminum than of cast iron. This may be a peculiar method of slitting the piston, or it may be casting in or bolting on pieces of steel to act as expansion controls. It may propose rings of special types or a peculiar location of the piston-pin, but the object is almost always the same; to provide an aluminum piston that will not slap when cold and will not stick when hot. A few of these schemes have been commercially successful and have played a major part in the rapid growth in the use of aluminum pistons.

Another factor in this development that has played an equal part in bringing the aluminum piston into prominence is the art of casting aluminum alloys in permanent iron molds. This process was imported from France in 1912 in a crude state. The first piston castings by this process were produced in 1915. The growth of the resulting industry was slow for a few years, but as the technique advanced it became very rapid. Since 1920 between 25,000,000 and 30,000,000 aluminum pistons have been made by this process. Its advantages over sand casting are many. The metal is hard and fine grained, free from porosity and from blowholes, and has excellent machining properties. Great numbers of castings may be duplicated almost exactly as to both dimensions and weight.

Another development in the last 3 or 4 years is the art of heat-treating aluminum pistons. By heat-treatment of permanent-mold pistons we are able greatly to relieve the casting strains which cause distortion in operation, and at the same time to increase the hardness to as much as 160 Brinell, thus enabling the piston to resist wear of the ring groove and skirt to a great extent.

## The Design of Aluminum Pistons

The most familiar design of aluminum piston, the one that brought it into prominence, is the so-called slit-skirt design. In this type of piston we have two bearing-faces of approximately 90 deg. each, separated from the head and ring section by horizontal slots. Up the center of one of these bearing faces, from the open end of the piston to the horizontal slot, runs a vertical slot. The purpose of the horizontal slots is to separate the bearing faces from the piston-head, preventing a direct flow of heat from the head to the skirt and thus maintaining a lower skirt-temperature. The purpose of the vertical slot is to allow the piston-wall to deflect slightly after it reaches the size of the cylinder bore, thus eliminating the possibility of seizure at high speeds. This is the design that has been in general use for about 7 years. The clearance provided with this type of piston is usually about the same as with cast iron.

Recently, particularly during the last year, a considerable demand has developed for still closer piston-clearance. There has been what A. Ludlow Clayden has referred to as a "piston-slap complex." It seems that the whole trade has concentrated on the elimination of piston slap and it sometimes seems that every noise occurring in an engine is referred to as piston slap. Be that as it may, the demand has been for closer clearances, and to meet this demand the insert piston has been developed.

This type is identical with the split-skirt type in most of its essential features. It has the two 90-deg. bearing-faces separated from the heat by horizontal slots. The piston-pin bosses are attached to the head exactly as in the split-skirt type. The connection from the piston-pin bosses to the edges of the bearing faces, however, instead of being by aluminum ribs, is by steel ribs cast into the piston-pin bosses and into the edges of the bearing faces, connecting the adjacent edges of the two bearing-faces in the same manner as do the aluminum ribs in the split-skirt piston. These two steel inserts carry the thrust loads of the piston against the cylinder-wall and act as spacers between the two bearing-faces, controlling the expansion across the bearing faces due to temperature. If the inserts are made of cold-rolled steel the permissible clearance will be about the same as with a cast-iron piston of like diameter. If they are made of Invar, a steel containing about 35 per cent nickel, which has practically no expansion in the range of temperature encountered in piston operation, the clearance may be less than is ordinarily employed with cast iron. Pistons of this type are heavier and somewhat more costly to produce than the all-aluminum type. They offer some problems in casting strain and distortion, incident to combining two materials of radically different coefficients of expansion in the same rigid structure, but they will operate with very small clearance.

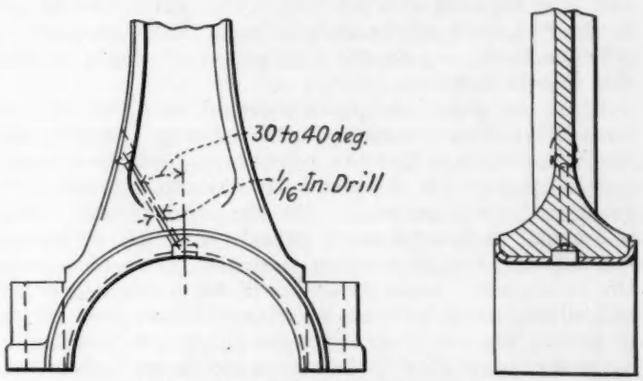


Fig. 1—Oil-Hole in Connecting-Rod
A Hole Is Drilled To Lead Oil from a Hole in the Crankpin to the Side of the Connecting-Rod on the Thrust Side of the Engine, for Lubrication of the Cylinder-Wall of Pressure-Lubricated Engines during the First Minutes of Running

## Installation of Aluminum Pistons

Certain things must be given attention in making any aluminum-piston installation. Chief among them is lubrication, particularly in cold weather. There has been much criticism of the aluminum piston because of what is known as scuffing. That ailment was a mystery for a long time, but it has now been thoroughly demonstrated that it is due to lubrication failure. It is never encountered in splash systems because in this case the dippers start throwing oil to the cylinder-walls immediately upon starting, almost regardless of oil temperature. The most aggravated condition is to be found in engines employing a full-pressure system with the oil-pump located high on the engine, drawing its supply through a small tube extending into the oil-pan. Oil cannot be drawn through such a tube at temperatures of 0 or even 20 deg. fahr.; therefore pistons and cylinders receive no fresh oil for several minutes after starting. Meantime the cold cylinder-walls are condensing water from the products of combustion, washing off the little oil remaining on the walls. I have seen enough water condensed in this way by a 2-min. run to drip from the crankshaft when the oil-pan was removed. This condensation is largely responsible for the rusting of cylinders and crankcases in winter. Such a condition is ruinous to any piston.

The cure for this trouble lies in providing means for getting fresh oil to the cylinder-walls as soon as the

engine is started, to maintain the film that is being destroyed by the condensing water-vapor. The best method that has been developed for this purpose, one that is being generally adopted by manufacturers using both cast-iron and aluminum pistons, is to drill a hole of about 1/16-in. diameter through the center of the connecting-rod bearing, as shown in Fig. 1, to register with a hole in the crankpin about 15 deg. from the top of the bearing. This hole should emerge on the outside of the rod on the thrust side of the engine near where the shank of the rod joins the big-end bearing. This will allow a few drops of oil to reach the outside of the rod to be thrown to the cylinder-walls immediately upon starting. In addition, the oil-pump should be in the oil-pan, all oil channels should be of generous size and as straight as possible, and the screen which is placed over the pump inlet should be of coarse enough mesh so oil can pass through it when cold. The screen should not be finer than 30 mesh; 20 is better. These precautions will practically assure the elimination of piston scuffing due to cold starting.

Rings on aluminum pistons should be fitted with at least 0.00125-in. clearance in the grooves. If they are much tighter than this the grooves are likely to contract on the rings when the piston is cold and prevent them from contacting properly with the cylinder-wall. This will result in several sharp reports when the engine is started, due to gas rushing past the piston-rings into the crankcase. These reports will be intermittent and will disappear quickly as the rings become free in the grooves. The condition is not dangerous but it is highly annoying, particularly when its cause is not understood.

It is not necessary to use bronze piston-pin bushings with aluminum pistons as the bearing conditions between the hardened-steel piston-pin and the aluminum piston are very satisfactory. The few manufacturers who have used piston-pin bushings have discarded them in favor of the aluminum bearing.

### Magnesium-Alloy Pistons

One other question is occasionally brought up about which I shall say a few words in closing. What is the probable future of magnesium as a piston material? Magnesium has some advantages over aluminum, of the same nature as those that aluminum has over cast iron, but not to the same degree. It has no advantage in thermal conductivity, but it has a considerable advantage in specific gravity. A magnesium piston can be made from 30 to 40 per cent lighter than an aluminum piston. However, the process for reducing the metal is not nearly so old nor so highly perfected as is that for aluminum, and the price is consequently much higher. The same is true of the art of casting. Nevertheless, the engineering world, always looking for the optimum results, recognizes the possibilities of magnesium as a piston material and there has already been one instance where the metal has been tried out on a fairly large commercial scale. Considering that this was the first commercial installation, it gave a good account of itself, but it was discarded in favor of aluminum for much the same reasons that caused the early aluminum-piston installations to be discarded, because the price is high and the designs and technique necessary for large-scale production are not yet thoroughly worked out. If the reward is sufficient the difficulties will be overcome. The fundamental advantages of lighter weight and high heat-conductivity are there and means will be found to realize them.

[2] M.S.A.E.—Engineer, manufacturers service division, Vacuum Oil Co., Buffalo.

[3] M.S.A.E.—Passenger-car engineer, Pierce-Arrow Motor Car Co., Buffalo.

[4] M.S.A.E.—Engineering department, Pierce-Arrow Motor Car Co., Buffalo.

### The Discussion

E. W. Kimball[2]:—Aluminum pistons have interested me for a number of years, and my initial experience with them was in 1914. The first aluminum pistons we used were sand cast and, as very little was known about machining to get the desired results, our experience was such as to condemn them. In 1916, when we put aluminum pistons into engines and sent them out to the public, we began to receive many complaints of piston slaps. The service department, with which I was then connected, was called upon to find a remedy, and a method of slotting the skirts was developed. This did not materially improve the condition, and we adopted a heat-treating and expansion method. At right angles to the piston-pin at the lower part of the skirt we inserted an expander, then the piston was heated by blow-torches at opposite sides until the skirt was expanded, after which it was chilled by dropping into a bucket of water. By fitting the piston tight, disregarding the conventional clearance of about 0.001 in. per in. of cylinder bore, with a bearing surface from the bottom of the skirt about 7/8 in. up the side, we eliminated about three-quarters of the piston slaps in that particular model. Then we began to get better pistons, die cast, through cooperation with the aluminum manufacturer, and they gave good results. In fact, I think that this same car is the only one in America that has used aluminum pistons exclusively in regular production since 1916, except for about 2 months. Perhaps it is unfortunate that the oil refiners do not have something to say in the designing of engines. If they did there would be several changes. It is true that most of the wear on both aluminum and cast-iron pistons, on the crankshaft, camshaft and valve-lifters, in fact on all the moving parts of the engine, occurs immediately after the engine is started. Particularly in the winter an oil is needed that has a low cold-test and still has high lubricating-value.

Charles Pleuthner[3]:—Iimprovements in lubrication, crankcase ventilation, oil filtering, exhaust-heated manifolds, carburetion, and other engine details have had an indirect bearing on the success of aluminum pistons. These are all features that contribute to the eventual success of the aluminum piston.

### The Problem of Piston Slap

Chairman P. B. Jackson[4]:—Until I made an investigation of piston slap recently I did not realize that in one engine-revolution the piston goes from one side of the cylinder to the other four times, resulting from both the gas-pressure and inertia forces. With an aluminum piston the inertia forces are more than the gas-pressure forces at 3500 r.p.m, and there are times when both act in the same direction.

How small does Mr. Welty think the clearance must be made, regardless of material, to reduce piston slap to a negligible amount? Also, what proportion of the heat that is not converted into work is taken out of the engine through the piston-rings; does more heat pass out of the piston through the rings than by radiation to the oil in the crankcase?

G. D. Welty:—It is true that the piston, according to theory at least, does oscillate back and forth from one side of the cylinder to the other several times during

each revolution. However, it is my belief that only the oscillation that occurs when the piston comes up against compression and is snapped over against the opposite side of the cylinder as the crankshaft goes over center at the time of the explosion results in a piston slap. That belief is based on a number of experiments I made in a laboratory by taking all of the pistons except one out of an engine and fitting that one with 0.008 to 0.010-in. clearance so that it would readily slap. Under these conditions there were half as many slaps as there were revolutions of the engine. That implies that the engine was operated at such slow speed that the slaps could be counted and compared with the number of revolutions. As the speed increased there still was one slap to two engine-revolutions until the speed became so high that the slaps could no longer be counted.

Mr. Jackson said that at high speeds the inertia forces are even greater than the gas-pressure forces. I doubt whether that applies to piston slap, because the piston is hot when running at high speed and the clearance is reduced by expansion, so there should be no slap at high speed.

I have no direct measurements of the comparative amounts, but much more heat is carried out through the rings and the skirt into the cylinder-walls and thence into the cooling water than is radiated to the crankcase oil. An attempt has been made by casting radiating fins on the inside of a piston to maintain a cooler piston-head so that higher compression could be used. We were successful in casting the fins to provide greatly increased radiating-surface, but the results obtained from the experiments with pistons of this type did not justify the manufacturer in adopting them. Very little difference was reported. On the other hand, there is no question that a large amount of heat is dissipated through the rings and skirt. An experiment has been performed by bolting a wooden wearing-shoe to an aluminum head. Temperatures sufficient to fuse the head partially were developed, showing that under ordinary conditions a large amount of heat must be dissipated through the skirt, A considerable quantity is also transferred through the rings into the cylinder-wall.

QUESTION:—What prevents the development of an aluminum piston-ring with a steel expander for light weight and superior heat-dissipation?

### Piston-Rings Discussed

MR. WELTY:—One of the things that would hold that development back is that it would be rather expensive to furnish twice as many pieces as now. Also, the room for a spring back of a ring is not great. The depth of the ring grooves would need to be increased and that would require more stock in the piston.

I do not know how successful an aluminum piston-ring would be. There would be the question of its retaining its temper under working conditions. Cast-iron piston-rings are highly specialized parts, all individually cast. I doubt whether we could successfully cast an aluminum piston-ring by the permanent-mold process. The section might be too thin. Rings would probably need to be cut from aluminum forgings.

PERCY BEST[5]:—What are the best widths of rings for pistons of various diameters, considering good functioning and low ring-pressure on the cylinder-walls?

MR. WELTY:—The tendency is to use narrower rings with aluminum pistons. It is almost universal practice on passenger-car engines up to 3½-in. bore to use four ⅛-in. rings. The use of four rings permits the use of an oil-control ring and still leaves three compression-rings. With narrower rings and grooves there is little tendency for the rings to tighten in the grooves in cold weather, and narrower rings will more closely follow slight imperfections in the cylinder bore. The ⅛-in. ring is being used on some 4 and 4¼-in. pistons; in fact, almost everywhere except in motorcoach pistons, where the 3/16-in. ring is still generally used. The service of a motorcoach engine is considered so severe that manufacturers hesitate to use the narrower rings.

### Aluminum Pistons for Diesel Engines

QUESTION:—How far have aluminum pistons for Diesel engines advanced?

MR. WELTY:—Mr. Jackson has had more recent experience than I as to that.

CHAIRMAN JACKSON:—Diesel-engine pistons are so much larger than those of automobile engines that the effect of the coefficient of expansion is far beyond anything that the automobile engineer has to consider. Also, operating temperatures vary more in a Diesel engine, locomotives sometimes running continuously at full throttle with water temperatures as low as 40 to 45 deg. fahr., while sometimes the water will be at 212 deg. when the engine is running at part throttle. These temperature conditions make the problem of expansion very serious in 9 to 12-in.-diameter pistons.

The Invar-steel struts described by Mr. Welty control the diameter at right angles to the piston-pin but not in a direction parallel to it. In small pistons this seems not to matter, but in large pistons it becomes a serious factor.

Diesel engineers desire to use aluminum pistons because they will make higher piston-temperatures possible, but I know only of the double-acting engine in the Ford factory that has been run successfully with an aluminum piston, and as this engine has a cross-head there is no trouble with piston slap.

Work is being done on pistons for the Ingersoll-Rand locomotive-engine, trying to use smaller clearances with aluminum than with cast iron. In the near future a composite piston will be developed for Diesel engines, using an aluminum head with either a cast-iron skirt or a skirt with controlled diameter.

MR. KIMBALL:—Will Mr. Welty explain a little more in detail the crankcase explosions he mentioned and state what ignites the gas in the crankcase?

MR. WELTY:—The gas apparently is not ignited in the crankcase but rushes past the rings when they are not in contact with the cylinder-wall because of binding from being fitted too tightly in the grooves. Once past the rings the gas escapes immediately into the crankcase through the openings in the piston around the piston-pin bosses. It is the rush of gas that causes the noise. The phenomenon is rarely observed with trunk-type pistons where the gas has no ready escape into the crankcase even if it does pass the rings.

QUESTION:—At what temperature will a magnesium piston melt? I have seen some pistons that were badly pitted. I do not know whether from the temperature or from the gas.

MR. WELTY:—A magnesium piston will fail at a somewhat lower temperature than an aluminum piston. This is below the fusion temperature of either material because both become soft before they fuse. However, the temperature at which either aluminum or magnesium will soften is far above any temperature that could be

[5] M.S.A.E.—Automobile designer, Pierce-Arrow Motor Car Co., Buffalo.

encountered in an engine except under some quite abnormal conditions.

### Pitting of Magnesium

CHAIRMAN JACKSON:—About 18 months ago we fitted a set of magnesium pistons in an engine. After some running, the cylinder-head was removed and the piston heads presented a surface so much pitted that they looked like relief maps of a very mountainous country. It was discovered that there was a considerable water-leak directly into the combustion-chamber. The trouble with these pistons was attributed to the water leak, as magnesium pistons have functioned satisfactorily in aircraft engines for periods equivalent to flights of 70,000 to 80,000 miles.

A MEMBER:—I have noticed the same on magnesium brake-shoes that we use. Would aluminum act the same?

MR. WELTY:—No, magnesium is a trickier material than aluminum. Under oxidizing conditions magnesium alloys convert quite readily to a sort of white powder that you may have observed around these cavities.

A. D. CAMP[6]:—Magnesium is susceptible to water corrosion but, under such conditions as prevail in an automobile engine, magnesium pistons should corrode no more than those of aluminum, even if water should find its way into the cylinders. The film of oil present acts as a protective coating. Failure of magnesium pistons in the past can be ascribed more to faulty design or the selection of an improper magnesium-alloy than to corrosion.

A MEMBER:—Has the speaker had any experience with aluminum pistons and alcohol fuel, or with step-top aluminum pistons and kerosene fuel?

MR. WELTY:—I am fairly familiar with the theory of the step-top piston. Several laboratories in this Country are working intensively on that, but we have done no work on it in our own laboratory.

### Locking the Piston-Pin

MR. KIMBALL:—What are the relative advantages of locking the piston-pin in the piston boss or in the small end of the connecting-rod?

MR. WELTY:—There are three practices: (*a*) locking the pin in the piston and floating it in the rod, (*b*) locking the pin in the rod and floating it in the piston, and (*c*) floating the pin in both and using lock wires in the piston to prevent endwise motion of the pin. The design using the lock wires is the best where it is possible. The pin may then be fitted quite tight in the piston by heating the piston in warm water and assembling with the rod and pin while the piston is warm. When the piston cools the pin will be quite tight in it and the oscillation upon starting the engine will be between the piston-pin and its bearing in the connecting-rod. After the piston is warm the bearing between the pin and the piston-pin boss will become free and the pin can then oscillate in the piston. This construction also makes possible a better design of the small end of the connecting-rod, particularly for an aluminum connecting-rod, because it does away with the clamping device and is much stronger. Of the other two practices I prefer locking the pin in the rod. In this case, the piston-pin should fit the piston with about 0.0005-in. clearance at room temperature.

[6] Research engineer, United States Aluminum Co., Buffalo.

# COAL A RAW MATERIAL, NOT A FUEL

LEADING gas-engineers are setting about to disprove what has long been considered an almost axiomatic theory, that coal is fuel, according to a recent report of the American Gas Association. The contention is that coal, which must for thousands of years remain the chief energy resource of the world, is an essential raw material from the social, scientific and industrial viewpoints.

These engineers feel that but a relatively few years will elapse before their ideas are proved. Basing their conclusions on the fact that from 1815 to 1914 the white population of the world increased threefold, while the production of mineral fuels, coal, oil and natural gas, increased 75 times, they foresee a rapid culmination of the theory that the burning of raw coal defeats conservation and at the same time is uneconomic and injurious to health.

The report states in part:

> Giant gas-plants situated at strategic points throughout the Country where coal is easily available, and a network of gas mains distributing clean gas at high pressures over long distances, is an accomplishment of the future. Already sufficient knowledge is at hand to prove the practicability of this endeavor. Two gas mains from Ohio to West Virginia, 300 miles long, distribute natural gas at relatively high pressure. Another natural-gas line, now under way, will be 450 miles long and will pass gas at 450 lb. pressure.

The situation in the Ruhr district in Germany is especially bright, these engineers feel. Throughout the great iron and coal district of the Ruhr, there are enormous coke-oven systems, resulting in the production of huge quantities of coke-oven gas. The proposed method of utilizing this gas, which is a by-product of the coke ovens, is to construct a main 450 miles long to transmit gas at 450-lb. pressure. The gas would be piped to the local gas companies serving towns and cities far away.

It is claimed in the report that

> While the German situation is comparable to that in certain sections of America, a furtherance of the idea will prove to be much more practical. The erection of giant gas-manufacturing plants, at the mine mouth or strategic points, will take the place of the German coke ovens. The coke that is made will then be utilized in the making of additional gas, by a process that has been proved economically sound for many years, with the result that these plants will be able to pipe large quantities of gas to the surrounding territories.
>
> There will be absolutely no waste attached to these huge gas-plants. The utilization of the coke for additional gas-making will solve the problem of its disposal in sections where there are huge batteries of coke ovens today. The other by-products of gas manufacture can be disposed of easily. These include valuable fertilizers, perfumes, tar and the like.

The gradual exhaustion of the natural-gas supply is a stimulus to the development of huge gas-plants. Already research is underway at the American Gas Association testing laboratory to determine the most economic means of mixing manufactured with the natural product.—*Gas Progress.*

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# Service Aviation, Aeronautical Engineering and Commercial Aviation

*By* EDWARD P. WARNER[1]

METROPOLITAN SECTION PAPER

INFLUENCE that the research and development work done in aeronautics by the naval and military services has had in the advancement of design and construction of airplanes and aircraft engines suitable for commercial operations is pointed out and exemplified by citing a few instances of direct adaptability of military types of airplane to commercial uses. Nearly all of this work would have been done much later or not at all if the airplane had been purely a commercial vehicle, but the constructor for purely commercial purposes and the commercial operator have had the benefit of it. Major fundamentals, such as speed, safety, reliability and economy, are the same in both types of aviation; divergencies between the requirements for the two kinds of service begin to appear in materiel, personnel, or methods of operation only at a somewhat advanced stage of evolution.

Although it is inadvisable, as a rule, to combine military and civil functions in one design, the methods of design and construction created for the naval service are largely suitable for commercial purposes, as many elements that enter into the designer's work are independent of the use to which the finished machine is to be put. Progress grows out of specific knowledge, and the same procedure that suggests to the designer means for improving the performance of a military machine can point the way toward production of a safer and more comfortable passenger-transport airplane.

The author discusses the general question of the desirability of a specifically commercial type of airplane engine that shall be less refined and less expensive than the military types and indicates that safety, reliability and economy are objectives alike of commercial and military aviation. The problem of economy in first cost is tied up with economy of operation. By simple financial analysis it is shown that an engine which weighs 2 lb. per hp. and costs $24 per hp. is on a commercial parity with one weighing 3 lb. per hp. and costing less than $12 per hp. A 10-per cent increase in weight of the airplane structure will be justified only if accompanied by a 20-per cent decrease in total cost. The economic penalty on excess weight is too heavy at present to sanction any appreciable lowering, on behalf of air-transport operators, of the standards established by the naval and military services for the selection and working of materials used in airplane construction.

Since privately owned airplanes, on the other hand, are likely to be in the air relatively few hours per year and as first price rather than economy of operation and upkeep is decidedly a factor in their purchase, an undoubted need exists for rugged, reliable and cheap powerplants for such machines, even though the low price is obtained at considerable sacrifice of efficiency. This is the field of low power and small size.

IN considering the relations between naval and commercial flying, it may be well to start with a frank recognition that in matters of detail the contrasts are more obvious than the similarities. Aside from the necessity for the best possible performance and great maneuvering ability in some service aircraft, the exigencies of naval operations have brought into being certain highly specialized accessories, such as aircraft carriers with their arresting gears and catapults, which in turn exert an influence upon the lines followed in the development of aircraft and their individual parts. The time may possibly come when the stimulation of study of those very accessories can be reckoned as one of the greatest services that the Navy has rendered to the non-military operator; but in commercial operation the use of catapults and arresting gears to shrink landing-fields within the compass of a city roof-top is still in the realm of imagination, a remotely conceivable prospect of the future, while in sea-going aviation the use of similar devices is an immediate reality ever to be regarded.

Whether in materiel, in personnel or in methods of operation, however, divergencies between civil and military or naval operations begin to appear only at a somewhat advanced stage of evolution, and up to that point we can all go along together with mutual profit.

It is unnecessary to argue at length the debt that aviation owes to the developmental work carried on or encouraged by the armed services. A few examples will suffice, and, in offering them, military flying can be treated as a whole. There is no necessity here for making a distinction between the shares that the Army and the Navy have had in the work, much of which indeed has been divided between the two services so that a line of demarcation would be hard to identify with exactness; nor for distinguishing between those pieces of work which have been carried on entirely within the services, those which have been carried on by private industry with direct service encouragement, and those which owe their inception to the expectation on the part of their promoters that service orders sufficient to pay the development costs would be forthcoming when a satisfactory article had been brought into production. In each of these instances Government interest has been an important, often indeed a necessary, factor, and the final result comes to the commercial operator or to the constructor for the purely commercial market as clear gain, with the expense of the experimental stage already covered.

## SINGLE DESIGN UNSUITED FOR DOUBLE PURPOSE

That final result, useful in civilian flying, is not usually a completed aircraft. Service types of airplane are unlikely to be directly applicable to air-transport, although there are exceptions even to that rule. Recent experience has demonstrated the feasibility of using high-speed observation-machines for mail-carrying with but little change in the fuselage arrangement, and large patrol flying-boats such as find place in the naval service

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could be converted for oversea passenger-carrying without change except to the superstructure and the interior arrangement of the hull. Several well-known European flying-boat designs have indeed been made up alternatively in military and commercial models, differing in no other respects than those just mentioned. It is the general rule, however, that it is inadvisable to unite the military and non-military functions in one design. If one aim is of predominant importance in the attack on an engineering problem, all other ends likely in any degree to conflict with that one should be, not merely subordinated, but so far as possible dismissed from consideration entirely, so that full attention can be concentrated on the primary object. The absence of any present or prospective provision for a subsidy to commercial air-transport in this Country assures against the making of any effort to force the designer or user of commercial airplanes to give heed to their possible military or naval application. The time may come, as it has come for the merchant marine, when such duality of function can properly be considered to some extent, but commercial aviation now has its own problems, which are so important and so difficult as to command the fullest application of the energies of engineer and operator, whose attention should be free from distraction by any questions not purely commercial in their nature.

### War-Airplane Research Has Broad Application

Although the complete airplane as developed by or for the naval service is unlikely to be directly usable for transport or other commercial purposes, the methods of design and construction created for the one prove largely suitable also for the other. Many of the elements that enter into the designer's work are independent in their nature of the use to which the finished machine is to be put. Researches undertaken with the military airplane alone in mind are likely, in fact, to have a much broader application. A particular instance is found in the studies supported by the services with the object of determining the loads on aircraft structures in flight and the necessary structural design-factors for various kinds of flying.

The intense concentration of scientific and engineering talent which was brought to bear on aeronautic problems during and immediately after the war produced results of immeasurable value not only in aeronautics but in other branches of engineering as well. The theory of the design of tapered struts and that of the action of the long continuous beam carrying combined lateral and direct loads, to mention only two instances, were greatly advanced in completeness between the years 1912 and 1920. The progress made was primarily the fruit of those patriotic impulses which made students of structural theory and other sciences in all countries lay down the work in which they had been specializing and turn to studies capable of aiding in the production of an airplane that would be more efficient as a military instrument. The results of their labors now stand available not only for application in commercial aircraft but in some instances for the work of the civil engineer as well, since methods originally devised with special reference to airplane wing-spars have proved useful in the treatment of steel girders.

Referring more broadly to aeronautical science as a whole and not to structural theory alone, practically all of the work included in the 500 reports and technical notes of the National Advisory Committee for Aeronautics and in the 900 reports and memoranda of the British Aeronautical Research Committee, as well as in the corresponding documents in other countries, would have been done much later or not at all if the airplane had been purely a commercial vehicle; yet of all this vast accumulation of material not 2 per cent is solely or predominantly military in its application.

### Air Service Developed Instruments and Methods

Not only researches on theory, but new types of construction also, have profited by Army and Navy support. The all-metal airplane is used quite as much in commerce now as in military flying, perhaps more, but it is safe to say that it would be used much less than it is in every sort of operation had it not been that the threatening shortage of aircraft lumber during the war gave special point to studies of the replacement of wood by metal. Out of those studies, carried on with greatest vigor in the European countries where the shortage was most threatening, emerged those general rules for the suppression of secondary stresses to which metal construction in airplanes must still conform. From the large-scale research-work of the last 12 years there has come, too, an acquaintance with the detailed technique of handling the materials and forms that prove especially suited for aircraft work, and in some cases the very discovery of the materials themselves and of the details of their properties, as well as the creation of the industries producing them, is owing primarily to service aircraft-demands. The extensive work done on the characteristics of high-strength light alloys during the last 12 years is a case in point.

Another series of connections binding together all varieties of aerial operations is found by going back another step toward the foundations of the aeronautic art, to the tools employed by the research worker in attacking the various specific problems set before him. Accurate data are now obtainable on the performance of airplanes, including all their properties of stability and control as well as their speed and rate of climb, only because of the availability of instruments and methods originally devised for making acceptance trials of military machines and analyzing their maneuvering powers. Progress grows out of specific knowledge, and the same procedure that serves to suggest to the designer means for improving the maneuverability of a fighting-machine can be invoked to point the way toward the production of a passenger-transport machine displaying maximum steadiness in flight and offering maximum comfort to the occupants of the cabin. Such work as that done by the National Advisory Committee for Aeronautics is of the greatest value in this connection, and it would hardly have been possible to undertake it with continuing governmental support had it been necessary for the appeal for such support to rest on commercial grounds alone.

### Commercial Operators Use Military-Type Engines

That which is true of the airplane structure holds with even greater force for the powerplant, which is interchangeable as a complete unit between military and commercial service. There has been much discussion in years past of the desirability of a special design of commercial engine, and there is undoubted need of rugged and reliable but cheap and relatively inefficient powerplants to go into airplanes for private ownership. In the high-powered division, however, the desirability of a distinct school of non-military design is more than problematical, and whatever conclusion might be arrived at on academic grounds, the present fact is that civil aviation depends on engines devoted to the military market. There is not now in production anywhere in the world, so far as I am

aware, a single engine of more than 100-hp. output which was not designed and built with primary reference to sales for installation on military or naval airplanes. The modern air-cooled engine, on which commercial flying is coming to depend so largely both in America and abroad, is essentially the result of stimulation by the armed services, the first intensive work in that field in Europe during the war having been undertaken with reference to a specific type of military airplane and presumably with the ultimate adoption of the engine in three-engine transport-machines very far from anyone's mind. It is unnecessary to remark upon the amount of progress made in increasing the reliability and life of both air-cooled and water-cooled engines in the last few years or upon the peculiarly proud share that American engineers have had in this progress.

The general question of the desirability of a specifically commercial type of engine that differs in some degree from the military model is one which has been discussed extensively in very recent years. Assuming a commercial demand large enough to justify incurring the preliminary development expense on such a special engine, the answer to this question would necessarily depend on a decision on what is fundamental to the success of commercial flying, or specifically of commercial air-transport, and on the relative importance of the several fundamental qualities. On a previous occasion when I discussed before the Society the problems of commercial aviation, I listed[2] its major elements as five in number: Speed, safety, reliability, economy, and comfort. That classification still stands, and indeed, with the substitution of the more general terms "performance" and "convenience of arrangement" for "speed" and "comfort" respectively, it holds for military flying as well. It may seem a little surprising that economy is included as a military virtue, but, quite aside from the fact that economy in the procurement of military supplies is always desirable and to be sought for in itself, low first-cost symbolizes the possibility of quick and easy production in quantity in the event that this is made necessary by the demands of war.

**Government interest has been an important factor in aeronautical development, and the final result comes to the commercial operator or to the constructor for the purely commercial market as clear gain.**

**Service types of airplane are unlikely to be directly applicable to air-transport but researches, instruments of measurement, data, designs, materials, and methods of construction conducted or evolved by the Army and Navy have been of immeasurable value to civil aviation.**

**Fundamental major requirements for commercial airplanes are speed, safety, reliability, economy, and comfort, and the first four of these are also requirements of military airplanes.**

**The economic penalty on excess weight in commercial airplanes is too great at present to sanction any appreciable lowering of standards set by the naval and military services for the selection and working of materials.**

### Problem of a Cheaper Commercial Powerplant

One can hardly arrange in order of their importance the five qualities listed, for it is essential that a reasonably satisfactory performance be attained under each head. It is possible, however, to discuss the relative degree of our present advancement in the several respects and to consider where we should seek most vigorously to raise the standard. Three years ago when I addressed[3] the Metropolitan Section at an aeronautic meeting, I expressed the view that fear of the air acted as the most serious barrier to the spread of passenger transport by air and that efforts should be concentrated not only on making air travel safer but also on making people believe that it is safe. The object should be to dissociate the word "thrill" from flying and to convince the potential customer that air travel is not only safe and sane, but that it positively approaches being staid and monotonous.

The inbred reluctance against going up into the air persists, but the progress made in overcoming it in the last 3 years has been greater than I should have dared to hope at the beginning of that period. Safety has, it seems to me, given way to reliability and economy as the most serious barriers to a more rapid increase of public support. Reliability records are good, but they must be made much better and more nearly independent of unfavorable territory and weather conditions, and passenger rates must be lowered. Both of those conclusions have their bearing on the selection of the material.

Insofar as reliability is dependent on the airplane and its engine, there is no reason for departure from military standards. No commercial operator can possibly be more interested in the continued functioning of this engine than is the pilot who flies a land-type airplane away from a carrier and out over the sea. Neither for saving of money nor for saving of weight can the services afford to take appreciable added chances of engine failure during the progress of a flight.

It is in connection with economy that doubt begins to arise. May it not be that the modern aircraft-engine costs more than it is worth commercially, as compared with one of cheaper and heavier construction, with more cast iron and fewer duralumin and alloy-steel forgings machined all over? May it not be that the quest for maximum efficiency in flight leads to the adoption of methods of airplane construction that are unnecessarily elaborate and expensive and that are unable to justify themselves in a field where the goal is the showing of a balance in black instead of red at the end of a year?

The question of how much could be saved, either in the engine or in the airplane, but especially in the former, by adopting cheaper materials and simpler and cruder forms is somewhat speculative, as the answer depends on how large a production could be maintained. The question, however, of how much must be saved to justify such a change of practice can be answered specifically for any given set of conditions. There need be no dependence on guesswork or judgment.

The problem of how much it costs to carry an extra pound of weight throughout the life of the airplane, and of the sum which, subtracted from first cost, has a present worth equal to that of the gradually accumulating amount representing additional fuel and maintenance due to the added weight, lends itself to relatively simple financial analysis. The exact result depends on how many hours per year the machine is in the air, on the

[2] See The Journal, February, 1923, p. 155.
[3] See The Journal, April, 1924, p. 457.

life of the flying equipment and on the proportion to the total number of trips borne by trips on which a full load is carried. On the average, however, a 1-lb. reduction in engine weight is worth about $15 in first cost under present conditions, while a corresponding reduction in airplane-structure weight has a cash value of about $20 or a little more. Putting that into terms of unit weights and costs, it means that an engine which weighs 2 lb. per hp. and costs $24 per hp., which is a good average for modern airplane powerplants of moderate power, is on a commercial parity with one weighing 3 lb. per hp. and costing less than half as much. I believe there will be agreement that such an engine could scarcely be built except in very large production.

### Excess Weight Penalizes Lower Standard

On the airplane-structure side, $10 per lb. of weight may be taken as an average of cost for commercial airplanes in lots of 20 to 50 after the engineering and development costs have been absorbed. A 10-per cent increase in structural weight would be justified only if accompanied by a decrease in total cost of a little more than 20 per cent, or from $10.00 to about $7.50 per lb. in terms of weight. For the air-transport machine there seems to be solid economic ground for the remark, originally offered as a casual expression of opinion and without analytic backing, of a British designer of my acquaintance, that he would "spend a pound to save a pound any time." It seems that the economic penalty on excess weight is too heavy at the present time to sanction on economic grounds any appreciable lowering of the standards set by the naval and military services for the selection and working of materials. The use of "military" engines on airlines need not be explained away as an accident of convenient availability.

Such a conclusion must, to be sure, be subject to certain reservations. The calculation of the money value of weight-saving was based on certain specific assumptions, one of which was a substantial continuity of use of the equipment throughout the year. The results apply only to regular air-transport. Furthermore, a sufficient financial reserve to obviate worry over increased first cost, provided it brings a commensurate saving in operating expenses, is assumed. The best may be cheapest in the long run, but it does very little good to reiterate that fact to the man who has not, and cannot raise, the money to buy the best. When first cost is a really serious consideration, or in an operation in which use of the equipment is likely to be intermittent, the market for something low priced begins to appear. In such cases continued employment of surplus war-equipment instead of a more modern and more efficient product is clearly justified.

Since privately owned airplanes are likely to be in the air relatively few hours per year, and as price on its own account, unbalanced against upkeep charges, is decidedly a factor in their purchase in most instances, it is in these machines that the need for low price, even though obtained at a considerable expense in efficiency, is most clear. This is, of course, the field of low power and small size. The Army or Navy may find uses of their own for a low-powered engine, one of 100 hp. or less, and if they do they may be able to draw upon the results of design work initiated with a commercial aim, so making direct return to the services in exchange for the profit that commercial aviation has drawn from military experience in the past.

### Must Convince Public of Aviation's Reliability

Passing over various other developments such as methods of navigation and ground aids to air-transport, that have also been made in large part, at least, by or with the assistance of one or other of the Air Services, another phase of the relation between military and commercial aviation which is wholly outside the field of material and not directly connected with the work of the automotive engineer, except insofar as the engineer is himself directly interested in the progress of air-transport, is the psychological aspect.

A statement which has been made to the effect that it would have been better for public interest in aviation if airplanes had never been used in the war, is open to debate. Certainly the airplane would not have been developed nearly so far, and not nearly so much aeronautic research would have been carried on, if there had been no possibility of military use of the airplane, but it is true that the nature of military and naval flying and of the reports that are made on such flying are not always directly encouraging to public use of the airplane for passenger transport. This is only in part the necessary consequence of the nature of the work of the Air Services, in which some kinds of flying are done in which most citizens have no desire to participate. All who are especially concerned with service or commercial aviation should do what they can to indicate to the public that the important thing, whether in military or civilian flying, is not so much the sensation of the moment as it is the steady operations that are carried on day after day, month after month, year after year, and which have very little news value.

Not only in commercial flying, which goes on over one route day after day, but in the Service as well, it should be made plain to those who are not directly connected with aviation that the important feature is the work that goes on continuously, and that such performances as that which was recognized in the presentation of the Schiff Trophy to Captain Campbell for his performance in flying 840 hr. in regular Marine-Corps operation during the year with no damage of any kind to the airplane and, of course, no injury to anyone of the personnel, are of the real essence of Service operations. In proportion as we can convince the public that aviation goes on continuously, has a definite and continuing work to do and is important even when it does not get into the newspaper headlines, we shall be able to lead people to think of the airplane as something they can use for their own purposes, to bring them to increasing realization of the importance of commercial aviation and to take an increasing interest in its development and extension in the United States.

# The X-Ray Testing of Metals at Watertown Arsenal

*By* H. H. LESTER[1]

DETROIT SECTION PAPER *Illustrated with* DRAWINGS AND PHOTOGRAPHS

TESTS of metals with X-rays, as made at Watertown Arsenal, are of two classes: (*a*) radiographic tests in which photographic images of internal details of the gross structure are obtained and (*b*) diffraction tests in which images are obtained that may be interpreted to give information regarding details of micro-structure of the constituents in the metal. The present paper deals with tests in the first class. Diffraction tests will eventually result in steels that have better physical properties required for special applications in industry, but such improvement must be accompanied by elimination of defects in the gross structure of forgings and castings before the greatest utility of better steels can be realized.

Radiographic testing gives pictures of defects whereby the nature of the defects can be determined, but their causes must be sought by logical deductions from other information. Cast-steel ingots are too large for study by X-ray methods, and forgings are not suited to these tests because the rolling reduces the defects in the ingot to too small a size for detection. It is more important to understand the nature of defects in structural units which are familiar as steel castings, hence the tests are of fundamental importance in studies tending to improve the gross structures, and are of importance also in certain kinds of inspection testing.

The test method involves merely taking X-ray pictures and interpreting them. Blow-holes, cracks and sand inclusions in the metal are revealed as dark spots on the negative. The apparatus used and the method of making the photographs are illustrated and described. Cavities revealed by the negatives result from various causes, as by a gas or by shrinking of the metal while cooling and insufficient size or improper placing of heads or risers intended to supply additional metal in the casting to offset shrinkage.

Cracks are revealed by X-rays as a series, some of which may not come to the surface, and are therefore very difficult to remove completely by welding. It is often necessary as a means of preventing repetition of such defects in castings from a given pattern, to correct the casting design.

As the cost of making radiographs is from $1 to $5 per picture and often from 40 to 50 pictures are needed for study of a single large casting, the method is slow and expensive; but in some cases it is justified as a means of inspecting an individual casting to assure against a serious accident. It may be used as a routine method of inspection of many small types of castings or of larger castings of simple design. Its great value lies, however, in developing better technique in the making of castings through the study of a few pilot castings. The expense of the test is returned to the foundry, often many times, in the reduction of rejections.

This method of X-ray testing tells nothing of the physical properties of the metal except as these are affected by the presence of defects in the gross structure, but the elimination of these defects through study may result in the realization in the castings of the intrinsic strength of the metal. The true strength of a steel casting is governed to a greater extent by the presence and location of gross structural defects than by the intrinsic strength of the metal, which cannot be realized in the strength of the casting until gross defects of structure can be eliminated. Alloy-steel castings may be little better than plain carbon-steel castings, as their strength is governed by the presence of gross defects. It seems to be poor economy to put expensive alloy-steels into poorly made castings. It is more important to seek to improve manufacturing methods than to seek to improve the intrinsic properties of the metal of the casting.

SOME confusion regarding the purposes that the X-ray testing of metals accomplishes seems to exist in the minds of many who are not in direct contact with this work. As we have used them at Watertown Arsenal, X-ray tests are divided into two general classes: (*a*) radiographic tests in which X-rays are used to obtain photographic images of internal details of gross structure and (*b*) diffraction tests in which X-rays are used to obtain photographic images that may be interpreted to give information regarding details of micro-structures of the constituents in the metal. This paper will deal mostly with tests in class (*a*).

The use of X-rays for diffraction tests has developed to such an extent that it is recognized as a definite branch of metallography. It has given us new concepts of the micro-structures in metals. It has enabled us to understand much that previously was obscure as to the nature of the effects produced by thermal treatments of steel. It has clarified our ideas concerning the nature of the much-discussed steel constituents; martensite, troostite and sorbite. We have gained definite concepts of the manner of occurrence of elastic and plastic deformations. We can visualize, as we could not before, the atomic relationships involved whereby only a fraction of 1 per cent of carbon added to iron changes this relatively weak material to steel, our strongest structural material.

The eventual results from the studies that are being made in our own and other laboratories may be steels that are stronger and more resistant to abrasion and shock, and that will have other improved physical properties required for special applications in industry. We have recognized, however, that improvements in the properties of steel must be accompanied by other improvements in the quality of manufactured units; that is, the strength of a forged bar containing seams or microscopic cracks will be governed to a greater extent by these defects than by the intrinsic properties of the steel in the bar. Seams and cracks are details of gross structure.

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Probably a greater need exists at present for the elimination of structural defects than for the development of new steels. The need for better intrinsic properties will be postponed and the greatest utility of improved steels cannot be realized until such defects are eliminated.

### Pictures Show Nature of Defects

The elimination of structural defects requires, primarily, clear concepts of their natures and causes. No single experimental method will give the complete information desired. Radiographic testing gives pictures of the defects that serve, in most cases, to determine their natures. The causes must be sought by logical deductions from other information.

All steel is cast initially. Much of it is later forged, rolled or otherwise shaped. The rest is molded directly into the structural units familiar as steel castings. These castings may be studied by X-ray methods. Ingots are too large for present radiographic tests, and forgings are not suited to these tests because the defects have been reduced, by working the metal, to a size too small for detection. However, it is more important to understand the nature of defects in castings than in forgings, because it is in the casting that the defect is first formed. If the casting can be made free from defects, the forging will be free from most of its defects. Radiographic tests are of fundamental importance in studies tending to improve gross structures and are of importance also in certain kinds of inspection.

The test method involves merely taking X-ray pictures and interpreting them. It rests upon the fact that X-radiation penetrates matter that is opaque to visible light. Metals absorb the radiation very rapidly, as light is absorbed in a heavy fog, but if there is a blow-hole, a crack or a sand inclusion in the metal, the radiation that goes through the defect will be absorbed to a less extent than the radiation that goes through the adjacent sound metal. Consequently, if a photographic film is interposed to intercept the emergent beam after it has passed through the metal, regions of greater photographic density will appear on the negative after development. These black spots on the film are the images of the defects. It is necessary then to interpret them.

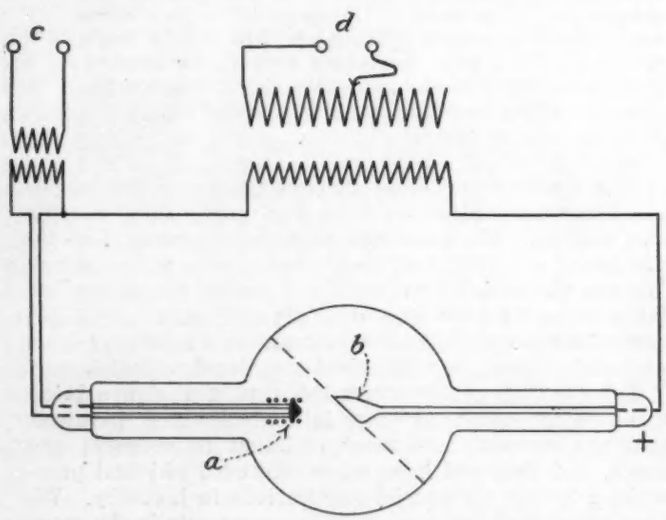


Fig. 1—X-Ray Tube and Electrical Connections
The Coolidge Tube Contains a Tungsten-Filament Cathode, *a*, and a Massive Tungsten-Anode, *b*. Electric Current from a Transformer, *c*, Heats the Filament, from Which the Heat Liberates Electrons. These Are Driven to the Anode, or Target, at Great Velocity by Rectified Potentials from a Second Transformer, *d*, Applied between the Cathode and the Anode. Absorption of the Energy of the Flying Electrons When They Strike the Target Produces X-Radiation, Which Goes Out in All Directions

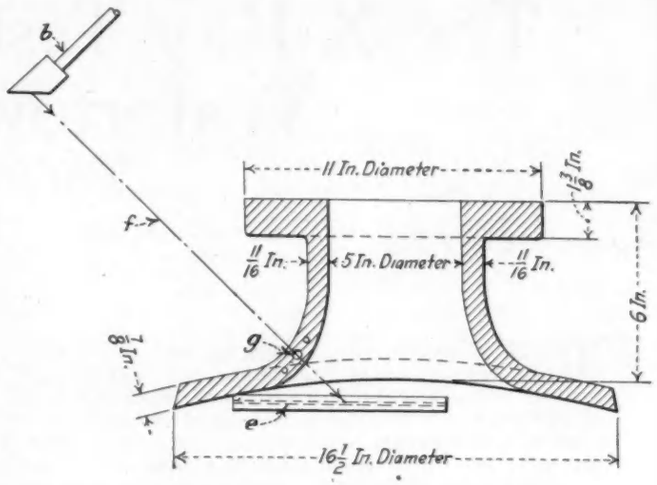


Fig. 2—Method of Making Radiographs
A Sensitized Film, *e*, Is Placed under the Steel Casting and X-Rays *f* from the Target *b* Are Directed through the Wall of the Casting upon the Film. Defects, Such as Cavities and Blow-Holes *g*, Permit the Rays To Pass More Readily than the Solid Metal and Are Revealed by Dark Spots on the Negative

### The Apparatus and Method of Use

The apparatus used is relatively simple. It consists essentially of an X-ray tube and apparatus for supplying necessary electrical energy to the tube. The tube used is the well known Coolidge X-ray tube developed by Dr. W. D. Coolidge in the General Electric Research Laboratories at Schenectady. The tube and a diagrammatic representation of the electrical connections are shown in Fig. 1. The tube consists of a highly evacuated chamber that contains a tungsten filament, *a*, which is the cathode of the tube, and a relatively massive piece of tungsten, placed at *b*, which is the anode. The filament is heated by electric current from a transformer, *c*. The hot filament liberates electrons which are driven to the anode, or target, as it is usually called, by rectified potentials from a transformer, *d*, applied between the hot-wire cathode and the massive tungsten-anode or target. Under the high potentials applied, the electrons acquire very great velocities and strike the target with considerable energy. The absorption of the energy of the flying electrons when they strike the target gives rise to X-radiation, which goes out in all directions from the area of impact on the face of the target. To obtain radiation of sufficient penetrating power to make pictures through about 3½ in. of steel, potentials of 200,000 to 250,000 volts are applied.

The method of using this apparatus for radiographing a casting is illustrated in Fig. 2, in which the sensitive film *e* is placed under the casting and X-rays *f* from the tungsten target *b* are directed through the wall of the casting upon the film, which, upon development, reveals the blow-holes *g* shown. A print from a typical negative taken in this way is shown in Fig. 3. The black spots are images of gas cavities. The casting in which they were present was cut open for further study and some of the cavities may be seen in Fig. 4, reproduced from a photograph of the cut section. While it was important to know that the cavities were present, it was more important for our purposes to know what caused them and how they could be prevented.

### Interpretation of Defects as to Causes

It is not the intent to present in this paper a lecture on foundry practice, but, to give a full understanding of these tests and their real usefulness, it is necessary to touch here and there on their applications in the foundry. Fig. 4 serves to illustrate how the radiographic tests are used. It was noticed that the cavities in the cut sections were for the most part bright-walled and for this reason must have been made by a reducing, rather than an oxidizing, gas. In addition, many of the cavities contained a black deposit which, on analysis, proved to be carbon and molding sand. It was evident, therefore, that the cavities were caused by the presence in the molten metal at the time of pouring of some molding-sand carrying a carbonaceous binder; that the carbonaceous material had broken down in the presence of the hot steel and formed carbon-monoxide gas, which left a solid residue of excess carbon and the silicon dioxide which were found in the cavities. After finding the cause of the trouble, the foundry engineers were able to devise a remedy.

Another typical casting-defect is shown in Fig. 5. In this case the cavities were caused by the shrinking of the metal at the time of solidification. Shrinkage cavities, which are well enough known, result from the fact that steel occupies less volume in its solid condition than when liquid. Since solidification starts at the surface, these cavities occur in the interior, which is the region last to solidify, to make up for the loss in volume. That castings may be free of such defects, it is the practice in foundries to cast, as integral parts of the casting, enlarged members which are last to solidify and hence should contain the shrinkage cavities. These members, called variously heads, sink heads or risers, are later cut off. The casting itself should be sound. Unfortunately, the risers do not always function as intended, due sometimes to incorrect dimensions, sometimes to incorrect placing, and sometimes to other causes. In such cases defects like those shown in Fig. 5 may occur. The remedy, as with the gas cavities, must be worked out by the foundry engineer.

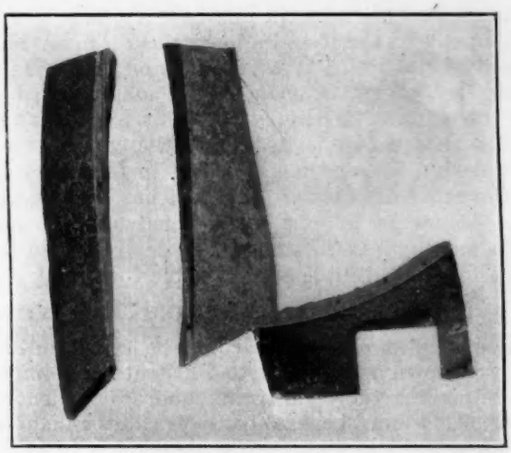

Fig. 4—Cut Section of Casting Exposing Cavities

When the Casting Was Cut on Sections through Defects Revealed by the Radiograph Reproduced in Fig. 3, the Cavities Exposed Were Bright-Walled, Which Indicated That They Were Caused by a Reducing Gas Rather Than an Oxidizing Gas. Some of the Cavities also Contained a Black Deposit. Upon Analysis, This Proved to Be Carbon and Molding Sand, from Which It Was Evident that the Cavities Were Caused by Presence in the Molten Metal of Some Molding Sand Containing a Carbonaceous Binder

Among the most difficult of defects to deal with are cracks like those illustrated in Fig. 6. The significant fact that radiography has revealed in connection with such defects is that there is usually not one crack but a series of cracks, some of which may come to the surface but many of which do not. It is the custom to weld cracks when they are observed. However, usually only one crack is seen on the surface and often only part of

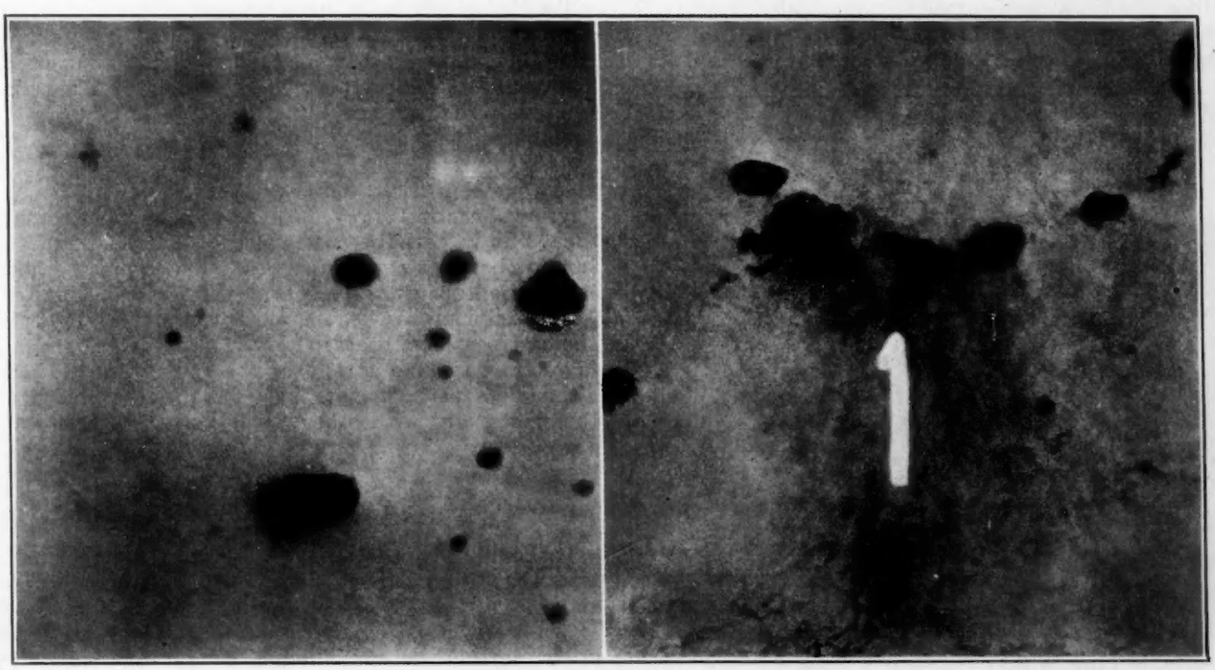

Fig. 3—Prints from Typical X-Ray Negatives

The Black Spots Are Images of Gas Cavities in a Casting for an 1100-Lb. Demolition Bomb To Be Carried by an Airplane. The Large Lighter Spot to the Upper Left of the White Figure 1 in the Right-Hand Picture Shows a Sand Inclusion. The Indistinct Markings at the Bottom Are Small Pipe-Cavities. Similar Defects in Subsequent Castings Were Eliminated by Manufacturing Technique. These Cavaties Are Shown in the Edges of the Cut Castings in Fig. 4

that one. For instance, in one case a crack was traced for a distance of about 15 in. by means of X-ray pictures, although only 4 in. of it was visible on the surface. In another case a system about 21 in. long showed only 9 in. on the surface. The only sure way to prevent repetition of such a defect is to remove the cause. Incorrect casting-design rather than mistakes in foundries is very often the cause of cracks. This fact has been recognized by the engineers of one large construction company, which is making an effort to modify designs of important castings to make them less susceptible to casting defects.

### Too Expensive for Routine Testing

Such examples of defects could be cited indefinitely but those shown will suffice to illustrate the important points in making the tests in the laboratory. There are, however, some more general considerations in connection with radiographic testing. The question is often asked, What is the greatest usefulness of the tests? As I see it, the radiography of cast metals cannot be a method of routine testing except in certain special cases because of the expense. It costs from $1 to $5 per picture, and often 40 or 50 pictures are required to study a casting. As a method of inspection it is slow; sometimes a week is required to inspect a single piece. Nevertheless, in some cases these considerations are counterbalanced by others that are more important. For instance, in one case a casting was to go into a service in which, if it failed, losses of several hundred thousand dollars might result. In another case castings were made to withstand 1200-lb. steam-pressure and the failure of a casting might result in an explosion that would wreck a $10,000,000 plant. The cost of the test may amount to many times that of the casting, but in such circumstances it still is economical. The cost is properly chargeable to insurance, or perhaps assurance. On the other hand, routine testing may often be applied as a method of inspection of many types of small casting or of larger castings of relatively simple design where the cost of testing is not excessive. Whether the tests can be made economically may be decided in most instances upon a study of the individual casting with reference to the service it is to render.

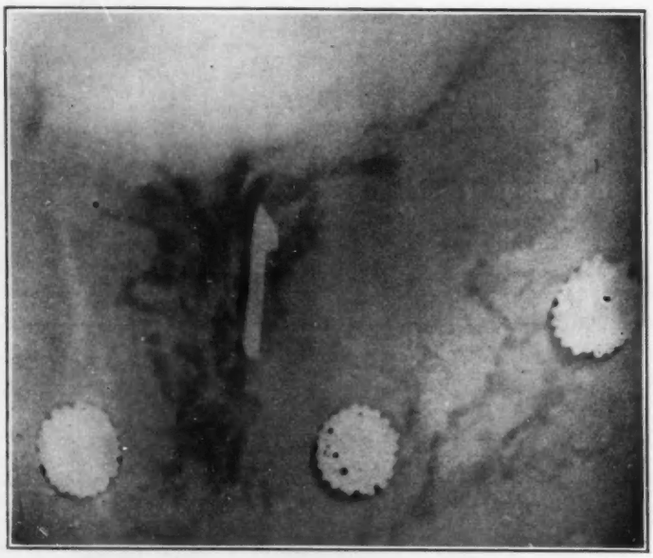

Fig. 5—Pipes in Bolt Flange of a Steel Casting
The Mottled Area at the White Figure 1 Is the Image of a Large Cavity, or Pipe, Caused by Shrinking of the Metal while Solidifying. Smaller Cavities Are Shown at the Lower Center. Such Defects Are Commonly Due to Incorrect Dimensions or Placing of Heads or Risers Intended To Supply Additional Molten Metal to the Interior of the Casting as It Cools

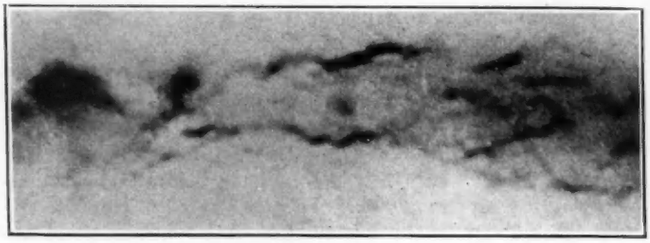

Fig. 6—System of Invisible Cracks Revealed by X-Rays
These Were in an Important Part of a Casting for a 16-In. Coast-Defense Gun and None of the Cracks Were Visible on the Surface. It Is Significant That Radiography Has Shown That Cracks Usually Occur as a Series and That Many Do Not Appear at the Surface, While Others That Do Show Are Often Concealed for a Large Part of Their Length, Hence Welding Does Not Always Fully Remedy the Defect

### Valuable for Developing Better Casting-Designs

The value of the method in developing manufacturing technique is more easily demonstrable. It has been the experience at Watertown that casting defects that are removed by correcting manufacturing technique tend not to recur. When the method is used to develop a proper method of manufacture by a study of a few pilot castings, the expense of the tests is returned to the foundry, often many times, in the reduction of rejections. At the same time, the consumer profits by receiving dependably sound material. Such use of the X-ray test-method is warranted only if enough castings are to be made from the same pattern to justify the expense of the development work. When a proper manufacturing method is developed for certain types of casting, there are likely to be returns on other castings that have not been subjected to X-ray study. This arises from the education of personnel as the result of intensive study of causes of defects in the castings that are examined. Some of the information obtained is transferred more or less unconsciously to the manufacture of material that is not subjected to the X-ray test. It was found, for instance, that notwithstanding a declining volume of production in the arsenal foundry, the cost per pound of making castings had been reduced steadily. This reduction in cost was due to a lessened number of rejections of castings that had not been subjected to development study but had undoubtedly profited from studies made on other castings.

A question that has been asked is, Have X-ray-developed castings proved their superiority in actual service? It is perhaps somewhat too early to be certain, but in the most conspicuous case where the method has been relied upon to give serviceable material, that of the 1200-lb. steam-pressure installation previously mentioned, the castings have been in service for 2 years and a leak is yet to be found in any of the radiographed castings, although considerable trouble has been experienced in the same plant with other castings that carry less pressure and for this reason were not subjected to X-ray tests.

### Defect-Free Castings Have Full Metal-Strength

It should be kept in mind in judging results from X-ray tests that this method tells nothing of the physical properties of the metal of the casting except as these properties are affected by the presence of cracks, cavities or other defects herein alluded to as details of gross structure. I mention this because some persons have had the impression that radiographic tests do give information as to the intrinsic strength of the casting. This impression is erroneous, or perhaps involves a cer-

tain confusion of ideas. If by careful study, either by X-rays or other methods, castings are developed that are free from gross defects, then the intrinsic strength of the metal may be realized in the strength of the casting. In this sense radiographic tests do give information as to the strength of the cast metal. There seems to be a tendency at present to feature alloy-steel castings. Such castings are desirable and necessary in some services but in many cases alloy steels have contained numerous shrinkage cavities, gas pockets, sand inclusions, and cracks. Such castings are no doubt somewhat stronger than if they had been made of plain carbon-steel, but not much stronger; the gross defects are the determining factors in the strength of such castings. It seems a waste of good material to put expensive alloys into poorly made castings.

Results obtained from studies of forged materials have not been encouraging. Defects in these materials are of a nature that renders them very difficult to detect by radiographic tests. Large cracks may be found but usually those most feared are not large. The forging operations close up blow-holes though the walls and the flattened-out cavities often are not welded.

### Higher Standard in Cast Steel Resulting

Besides the specific results, some general aspects of the application of X-ray inspection-tests are encouraging. Because of the emphasis that has been placed on internal gross defects, pressure has been brought on the foundries to eliminate them. This pressure has stimulated intensive study by other than X-ray methods. The consumer may obtain the X-ray picture and study it at leisure. It reveals defects in a way that he can appreciate. Usually he has no means of breaking or sawing castings for study, but he can take the X-ray results into a committee meeting for examination. Results that seem to be working out are that structural engineers are obtaining a better understanding of the problems of the foundryman and the foundries are learning how to make castings that meet the requirements of the engineer. A definitely higher standard of quality in cast steel has resulted from this cooperation and from intensive development-studies in the foundries. Perfection has not been reached, nor closely approached, but the outlook for further improvement is encouraging.

In a broad sense we feel that in this work we have been contributing toward the economic strength of the Country. In the markets of the world it will perhaps be increasingly difficult for us to compete successfully with other countries in finished products unless our manufactured material has an element of quality that is not found in competing products. Such studies as are being made in the attempt to improve cast metal cannot but be of definite value in this matter of quality.

### THE DISCUSSION

QUESTION:—Do not the X-rays have a harmful effect upon the operators?

DR. H. H. LESTER:—We keep the X-ray tube in a room lined with 12 tons of lead. The operator looks at the machine through a periscope. With proper protection the operator is not necessarily exposed to danger.

QUESTION:—What is the cost of an X-ray-testing apparatus?

DR. LESTER:—The X-ray apparatus itself costs about $8000, and auxiliary equipment about $2000, making an investment of $10,000, exclusive of the building.

QUESTION:—Can other metals than steel be X-rayed?

DR. LESTER:—Most of our work has been with steel, but so far as I know there is no essential difference in the technique for other metals. Some metals are more opaque to X-rays than others.

QUESTION:—Do you use the fluoroscope?

DR. LESTER:—No, it is difficult to use and can be used practically for thicknesses of metal not greater than ½ or ¾ in.

QUESTION:—Would it not be practical to use the fluoroscopic method on small castings like those used in an automobile?

DR. LESTER:—There is more hope for the fluoroscopic method in the case of automobile castings. Perhaps tires, insulation of wires and carbureter mechanisms can be so examined.

QUESTION:—To what extent do you use the X-ray method at the Watertown Arsenal?

DR. LESTER:—The latest rule is that every new casting must be subjected to the X-ray test. Formerly we X-rayed only every important one. We also examine all important castings purchased outside, if possible.

QUESTION:—Can this X-ray work be done anywhere?

DR. LESTER:—It can be done best in a room properly designed for the work. The equipment usually is too bulky to take into a shop. In some special cases that have arisen it was thought desirable to take the machine to the job, but the work was not done, largely because of the expense that would be involved.

QUESTION:—What has the Bureau of Standards done in this connection?

DR. LESTER:—The Bureau has a radiographic machine but I do not know what work has been done with it there.

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# MOTOR-VEHICLE EXPORTS INCREASE

AMONG the outstanding developments in the automotive industry is the fact that, in 1925, foreign countries bought more than 500,000 motor-vehicles, a gain of 39 per cent. over 1924. Exports of automobile passenger-cars from the United States alone during 1925, the last year for which complete official figures are available, amounted to 244,300. These had a total value of $184,900,534. During the same year 58,615 trucks and motorcoaches were exported having a total value of $37,703,250. In other words, this Country in 1925 exported 302,915 motor-vehicles having a total value of $22,603,784.

The Department of Commerce is authority for the statement that 24,564,574 motor-vehicles are registered in the world. Of this number, foreign lands now have but 19 per cent, but their demand for motor transport is growing rapidly. Motor-vehicle factories are busy all over the world. At the same time American motor-vehicle exports are increasing each month.

Vol. XXI August, 1927 No. 2

# Dynamometer Test of Brake-Drum Heat in Dual Wheels

*By* C. W. Bedford[1] and Ernest Blaker[2]

Transportation and Service Meeting Paper *Illustrated with* Photographs and Charts

PREMATURE failures in the summer of 1925 of inner-dual motorcoach tires and tubes on 20-in. wheels in which the brake-drum was directly under and close to the rim caused the rubber companies to undertake development work on tire beads, flaps and tubes; but no solution of the problem was to be found by a change of the rubber compound. The need for definite information regarding actual tire temperatures developed in road service led to extensive joint tests in Florida early in 1926 by a tire and a wheel company. In these tests a brake-drum temperature was maintained as nearly as possible constant at 475 deg. fahr. above atmospheric temperature by successively accelerating the car and applying the brake. When it was judged that this temperature had been reached, stops were made to take thermocouple readings of the temperature.

Realization that tests based on constant brake-drum temperature would not give as true a comparison of the heating tendency of different wheels and the effectiveness of heat-dissipating devices as tests based on constant brake-horsepower input, the B. F. Goodrich Co. undertook a series of tests with a Sprague dynamometer tire-testing machine. These were continued over a period of 3 months with apparatus described by the authors. Comparative tests were made on four different wheels, of which two were disc wheels, one 20 in. and the other 24 in. in diameter. The other two were steel-spoke wheels of different types but both 20 in. in diameter. Two ventilating fans were supplied with one of the latter wheels. Two cast-steel brake-drums, similar in other respects, but one having surface corrugations extending from the flange part way across the outside of the drum and the other having substituted for these corrugations a strip of Goodrich superheat packing 1/32 in. thick, were also used in making the tests.

The method of conducting the test and of taking temperature readings from the drum and the tire beads is described. In all tests 6-in. dual tires were mounted on the wheels and inflated to the standard pressure of 80 lb. To obtain comparative data, variables were introduced by running the wheels at various road speeds, with and without standard tire-load, with the different drums and with and without the fans fitted, and, in the case of one of the steel-spoke wheels, by mounting a fan separately from the wheel and directing its current on the drum at different angles.

Data derived from the tests and presented in tables and charts lead, among others, to the conclusions that (*a*) "burned beads" cannot result from internal heat due to tire flexing but must be attributed to brake-drum heat; (*b*) excessive flexing due to under-inflation, heavy loading or high speed will accentuate the effect of brake-drum heat by decreasing the temperature gradient between the beads and the tread; (*c*) increase in the speed of rotation for a given brake-horsepower input increases the cooling properties of a wheel; (*d*) for the heat-dissipating devices tested, substantially the same relative improvement in cooling effect on all wheels was noted; (*e*) forced ventilation by external fans is indicated as an aid for dissipation of brake-drum heat, and the slower the speed of a motorcoach is, the greater should be the effect; (*f*) on the basis of constant brake-horsepower the wheels show a wider temperature difference than on the basis of constant brake-drum temperature but in the same order; and (*g*) comparison on the basis of constant brake-horsepower more nearly reflects the relative cooling properties of the wheels in actual service.

PREMATURE failure of tires and tubes caused by brake-drum heat was first seriously encountered in the summer of 1925 and occurred with 20-in. inner-dual tires on wheels having the brake drum directly under and close to the rim. No real difficulty was encountered with 24-in. dual or 20-in. single tires, nor with dual tires used in light service with small drums or with the drums not directly under the rims.

The rubber companies at once undertook development work on tire beads, flaps, and tubes, with little prospect, however, of making rubber a refractory material. No solution of the problem is to be found by a change of rubber or compound.

Realizing the need for definite information regarding actual tire temperatures developed in road service, the Goodyear Tire & Rubber Co., through its highway engineer, G. M. Sprowls, and the Budd Wheel Co. conducted a series of extensive tests in Florida in the early part of 1926, and circulated reports on them. These tests were carried out by maintaining as nearly as possible a constant brake-drum temperature of 475 deg. fahr. above atmospheric temperature. This brake-drum temperature was reached and maintained by cycles of rapid acceleration followed by deceleration by brake application, with stops to take thermocouple readings whenever it was judged by experience that the desired temperature conditions had been reached.

It was clearly recognized from the first that tests based on constant brake-drum temperatures would not give as true a comparison of the different wheels or heat-dissipating devices as a test based upon constant brake-horsepower input. The difficulties of making road tests based on brake-horsepower as compared with the relative ease of testing on the basis of constant brake-drum temperature led to the adoption of the latter method, since it was evident that such a comparison would give valuable qualitative results.

Early in April, 1926, the attention of the B. F. Goodrich Co. was called to the fact that its large Sprague

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[2] Manager, temperature-control department, B. F. Goodrich Co., Akron, Ohio.

dynamometer testing-machine furnished a ready means of studying the effect of brake-drum heat on tire temperatures under known conditions of brake-horsepower input. Realizing the desirability of quantitative data, tests were started at once and continued over a period of 3 months. The data here presented are offered in the spirit of service. They are in no way contradictory of previous reports but rather an amplification, in that they are more nearly comparable with actual road-service on motorcoaches and present more clearly the actual conditions that exist.

The tests were performed on a special 25-hp. Sprague dynamometer tire-testing machine of 20,000-lb. capacity designed for testing truck tires, and were conducted under the personal direction of F. L. Haushalter, of the department of tests. The motor unit of the dynamometer, carrying the wheels and tires to be tested, as shown in Fig. 1, was mounted on a movable base whereby the tires were made to bear on a "road wheel" 1/300 mile in circumference. This road wheel in turn was attached to the shaft of the generator unit, which was mounted on a fixed base. A Toledo scale of 20,000-lb. capacity was used to give load reading. The road wheel had a 14-in. face with grooves 0.2 in. deep and 0.5 in. apart. This wheel width permitted the dual tires to run in full tread-contact. The hub carrying the brake drum and the dual wheels was mounted on a fixed axle provided with standard Timken roller bearings, and the drive was by floating axle from the motor. The brake was of standard cast-steel Westinghouse Air Brake Co. type, with steel shoes. A horizontal torque-arm was attached to the brake and carried a fine-threaded hand-screw for varying the braking effort. The free end of this torque-arm rested on a vertical support attached to a platform beam-scale of 500-lb. capacity. The torque-

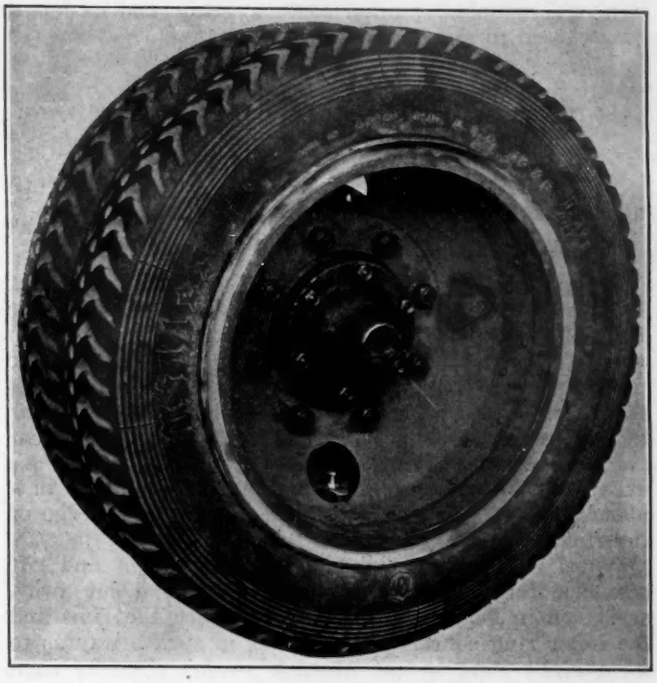

FIG. 2—WHEEL *A*, DUAL DISC TYPE FOR 20-IN. RIM, FITTED WITH 6-IN. TIRES

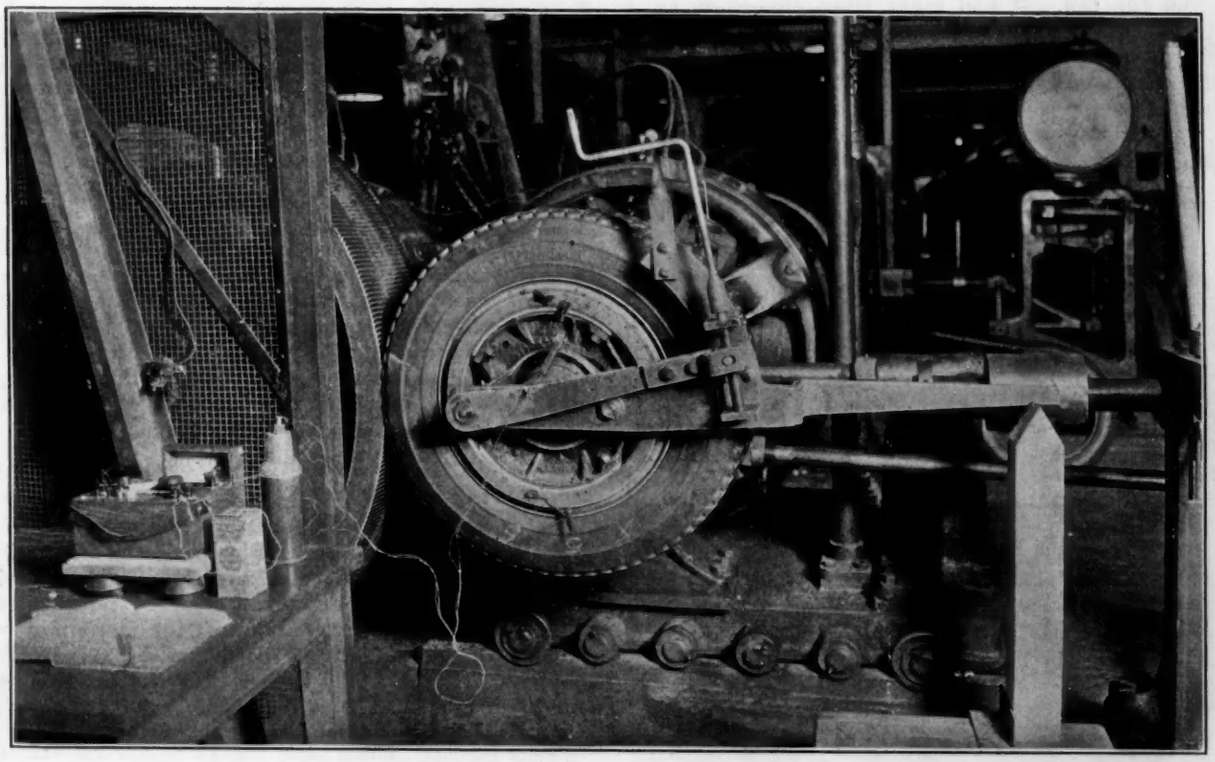

FIG. 1—SET-UP OF ELECTRIC-DYNAMOMETER TIRE-TESTING MACHINE

The Motor Unit of the Dynamometer, Carrying the Wheel and Tires To Be Tested, Is Mounted on a Movable Base So That the Tires Can Bear Against a "Road Wheel" 1/300 Mile in Circumference. This Wheel Is Attached to the Shaft of the Generator Unit, Which Is Mounted on a Fixed Base. A Toledo Scale of 20,000-Lb. Capacity Gives the Load Reading. The Hub Carrying the Brake Drum and Dual Wheel Was Mounted on a Fixed Axle Turning in Roller Bearings. The Drive Was by a Floating Axle from the Electric Motor. The Brake Was of Westinghouse Standard Cast-Steel Type with Steel Shoes. A Horizontal Torque-Arm Attached to This Carried a Fine-Threaded Handscrew That Provided Means for Securing Variations in the Braking Effort. The Free End of the Torque-Arm Rested on a Vertical Support Attached to a Platform Scale of 500-Lb. Capacity Which Is Partly Shown in the Lower Right Corner

arm was mounted in a horizontal position and its length was chosen to give the following constant:

$$\text{Hp.} = PN/1500$$

where

Hp. = horsepower developed in braking
$P$ = load on platform scale
$N$ = speed of wheel in r.p.m.

The tare factor of the brake was checked frequently and found to be practically constant.

### Wheels, Fans and Drums Used in Test

Comparative tests were made on four different wheels. Wheels *A* and *B*, of 20 and 24-in. diameter, shown in Figs. 2 and 3 respectively, were of standard dual right and left disc-pattern, which, when bolted together and to the standard hub, gave a Y-section, as in the cross-section in Fig. 4. They presented solid faces from hub to rim except for two small holes 180 deg. apart near the rims for projecting valve-stems. With the brake drum in position, the outer edge of the drum was in a plane about 1⅛ in. from the plane of the outer edge of the rim flange over the drum.

Wheel *C*, for 20-in. rims, was of the right and left disc-type for dual tires, but with the metal cut away, as shown in Fig. 5, in the parts between the rims and the inner ring containing bolt-holes in such a way as to form eight steel spokes in each section opposite each other. This wheel was interchangeable with wheels *A* and *B* on the standard hub, and with it were furnished two ventilating fans which could be bolted between the right and left discs so that the plates were opposite the openings or between the spokes. Fan. No. 1 had a steel band 1¾ in. wide and 20½ in. in diameter surrounding the fan. It had eight veins shaped so as to cause circulation of air over the brake drum when the wheel was in motion. Fan No. 2, shown in Fig. 6, had a 3-in. steel band of the same diameter as in fan No. 1 and which completely closed the space between the two rims. It had the same number of vanes but these were shaped so as to give increased circulation at the same

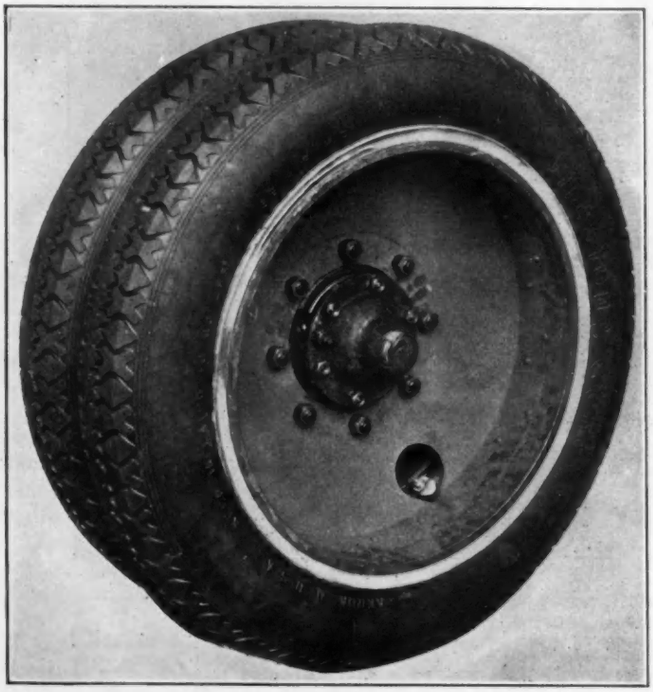

Fig. 3—Wheel *B*, Dual Disc Type for 24-In. Rim, Fitted with 6-In. Tires

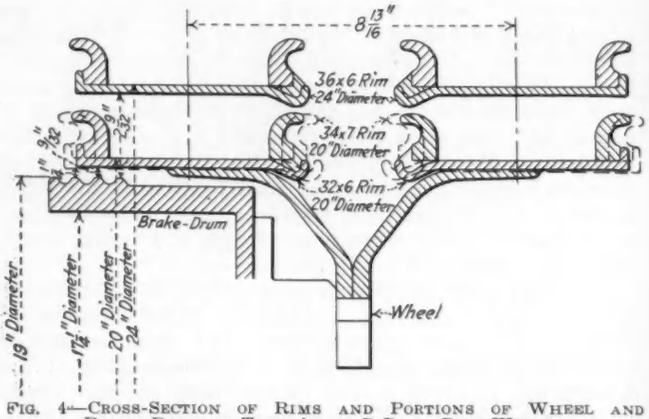


Fig. 4—Cross-Section of Rims and Portions of Wheel and Brake Drum of Type *A* and *B* Dual Disc Wheels

Clearance between the 20-In. Rim and a 17¼-In. Brake Drum Is ¼ In. at One Point and 9/32 In. at Another Point. With the 24-In. Rim This Clearance Is 2 9/32 In.

speed as compared with fan No. 1. These fans might have been added to wheels *A* and *B* but they could not be adapted to wheel *D*, and their use was therefore limited to wheel *C*.

Wheel *D*, for 20-in. rims, had a hub and eight spokes comprised in a malleable casting, as shown in Fig. 7. The standard rims for the dual wheels were mounted directly on the ends of the spokes. A deflector band, 20½ in. in diameter, completely filled the space between the two rims in the assembly. This deflector band was used in all tests of this wheel.

Two cast-steel brake-drums, designated drum No. 1, drum No. 2, were used. The former had the usual outer corrugated surface extending from the flange part way across the outside of the drum. Drum No. 2 was a similar drum but the corrugations were removed and replaced by a strip of Goodrich superheat packing 1/32 in. thick, held in place by thin wire. The objects of this change were to (*a*) increase the space between the drum and the rim, thereby permitting greater convection cooling, and (*b*) decrease the direct radiation from the drum to the rim.

Drum No. 1 was tested with each of the four wheels set up as received, that is, with a fan between the spoke discs of wheel *C* and with a deflector band between the rims of wheel *D*. Drum No. 2 was tested on the three 20-in. wheels combined with a steel deflector-band that completely closed the opening between the two rims of wheels *A* and *C* to make the tests comparable with wheel *D*, which contained this band as a standard part.

### Temperature Measurements and Method of Testing

Room temperatures in the neighborhood of the dynamometer were registered on a recording thermometer. Brake-drum temperatures and temperatures of the interface between the beads and rims were determined by thermocouples of copper-advance wire and a potentiometer indicator. Two pairs of thermocouples were mounted on the surface of the drums. The pairs were about 180 deg. apart and in each pair the junctions were separated by about 1 in. This was done for the purpose of obtaining check readings.

Three thermocouples were placed under the tire bead which was directly over the brake drum and which, on a motorcoach, would be called the outer bead of the inside tire. Two thermocouples were, as a rule, placed under that bead of the second tire which corresponds to the inner bead of the outer tire on a motorcoach.

In all tests 6-in. dual-tires were mounted on the wheels

and run under standard loading and inflation conditions. In tests run to constant brake-drum temperatures the braking effort was manipulated by a hand-screw to hold the brake-drum temperature constant until an equilibrium temperature was reached at the bead directly over the brake drum. Tests were then repeated, maintaining a constant brake-horsepower until a corresponding equilibrium temperature of both drum and bead was attained. The results of both types of test naturally fall on the same curve. Check runs were made in sufficient number to check any point which was found to be widely out of line, while results which varied only slightly from their apparent true value were recorded as originally obtained. In this way a series of drum and bead temperatures, with corresponding brake horsepower, were determined. All data as given in the accompanying tables and charts are corrected for room temperature and given as rise in degrees fahrenheit above atmospheric temperature.

This method of testing on the Sprague dynamometer

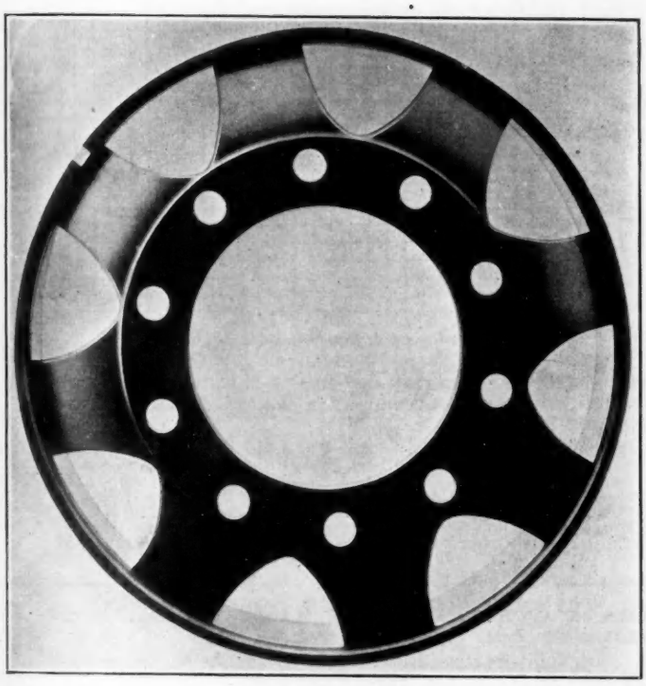

FIG. 5—ONE DISC OF WHEEL *C*, WITH RIM
Metal Is Cut from the Disc between the Rim and Inner Bolt-Hole Ring To Form Eight Spokes, Which Are Placed Opposite the Spokes in the Mating Disc in the Dual Assembly

cannot be said to duplicate operating conditions on the road in every respect. Road conditions, however, are so widely varied that the dynamometer test is greatly superior. Such factors as radiant heat from hot road-surfaces, variable winds, variations in acceleration or deceleration, effect of wheel housings, and personal equation of the driver are entirely eliminated or are maintained constant. This modification of road variables to constant conditions for laboratory-test purposes is believed to be one of the most favorable aspects of the Sprague-dynamometer test. Windage from front to rear of a motorcoach, varying with speed, was also present in the dynamometer test due to rotation of the large road-wheel against which the wheels and tires were run. The fanning action of this road-wheel was constant for a given speed. Room temperatures during the tests varied between 74 and 91 deg. fahr., and corrections were made at every reading of wheel temperatures.

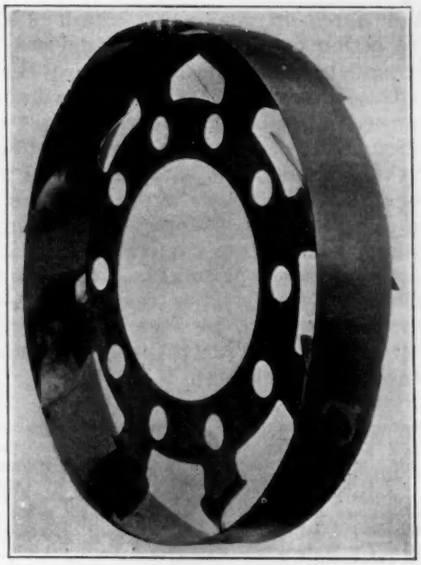

FIG. 6—VENTILATING FAN NO. 2 SUPPLIED WITH WHEEL *C*
This Has Eight Vanes Which Fit between the Spokes of the Wheel and Are Shaped To Cause the Air To Circulate over the Brake Drum When the Wheel Is in Motion. The Vanes Are Encircled by a 3-In. Steel Band That Completely Closes the Space between the Two Rims of a Dual Wheel. This Fan and Fan No. 1 Might Have Been Applied to Disc Wheels *A* and *B* but Could Not be Adapted to Wheel *D*, Hence Their Use Was Limited to Wheel *C*

## FACTORS INCLUDED IN CHARTS OF RESULTS

The dynamometer set-up for measuring brake horsepower and the resultant drum and bead temperatures permitted the inclusion of other factors than the constant brake-drum temperatures as used in previous road tests.

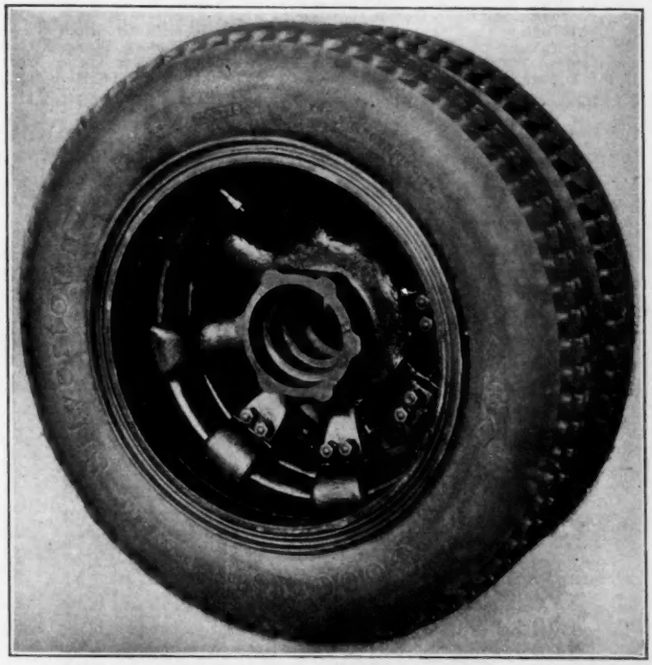

FIG. 7—DUAL WHEEL *D*, 20-IN. RIMS AND 6-IN. TIRES
The Hub and Eight Spokes Comprise a Malleable Casting on Which Standard Demountable Rims Are Mounted Directly on the Ends of the Spokes. A Deflector Band 20¼-In. in Diameter and Completely Filling the Space between the Two Rims Was Used in All Tests of This Wheel

The effect of speed on the cooling action of the wheels, the relative action of various cooling-devices with variable brake-horsepower, the effect of tire heat as such and the effect of external fans as auxiliary cooling-attachments were also included in the tests.

It would, therefore, be rather unwieldy to report the results of these tests by the same graphical methods as used by other investigators wherein only one point was obtained on a given curve. Moreover, the comparison of different wheels and cooling devices varies widely as the speed or brake horsepower increases or decreases. The complete picture of the result is, consequently, obtainable only by curves as shown in Figs. 8 to 15. Numerals placed on the individual curves refer to Tables 1 to 6 for location of numerical data.

The comparison of the four wheels as received, using standard drum No. 1, is shown in Fig. 8 at the left on the basis of brake horsepower versus bead temperature, and at the right on the basis of drum temperature versus bead temperature when tested at a speed of 30 m.p.h. It is noteworthy that the three 20-in. wheels maintained the same relative positions but quantitatively are farther apart on the brake-horsepower-bead-temperature curves than on the drum-temperature-bead-temperature curves. The relative order, however, of wheels *B* and *D* is reversed by the two methods of graphing the data. Conduction of heat from the drum to spokes, hub, axle, or other metal parts, as well as convection cooling of the brake drum itself, are two factors that are eliminated in any test based on constant brake-drum temperature, since

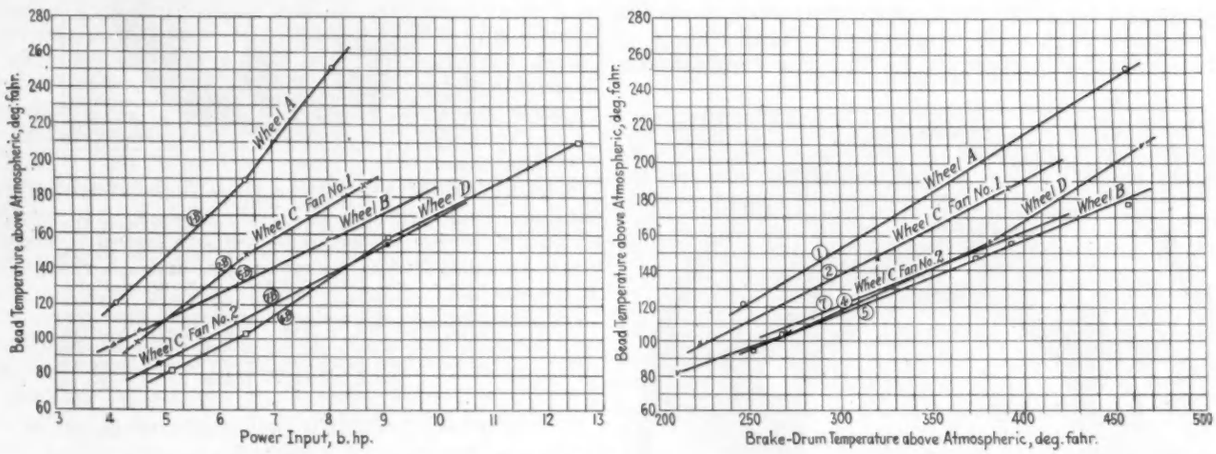


FIG. 8—COMPARISON OF HEATING OF FOUR WHEELS ON THE BASIS OF RELATION OF BRAKE-HORSEPOWER INPUT TO BEAD TEMPERATURE AND OF BRAKE-DRUM TEMPERATURE TO BEAD TEMPERATURE, AT A WHEEL SPEED EQUIVALENT TO A ROAD SPEED OF 30 M.P.H.

Curves in the Chart at the Left Show the Relation of Brake Horsepower to Bead Temperatures in Motorcoach Pneumatic Tires, and Those in the Chart at the Right Show the Relation of Brake-Drum Temperatures to Bead Temperatures. Letter *B* after the Numerals in the Circles Refers to Bead Temperature in Table 1. In All Cases the Wheels Were Fitted with Cast-Steel Brake-Drum No. 1 Having an Outer Corrugated Surface. In One Test Wheel *C* Was Fitted with Fan No. 1 Having Eight Vanes Encircled by a 1¾x20½-In. Steel Band, and in Another Test Fitted with Fan No. 2 Encircled by a 3x20½-In. Steel Band. Note That the Three 20-In. Wheels, *A*, *C* and *D*, Maintained the Same Relative Positions but Are Farther Apart Quantitively on the Brake-Horsepower Curves Than on the Drum-Temperature Curves. The Relative Order of Wheels *B* and *D*, However, Is Reversed by the Two Methods of Test

TABLE 1—TEMPERATURE RISE IN BRAKE DRUM AND TIRE BEADS AT A WHEEL SPEED OF 30 M.P.H. WITH DUAL TIRES LOADED TO 4400 LB. AND INFLATED TO 80-LB. PRESSURE

| Wheel | Curve, No. | Power Input, B.Hp. | Temperature Rise above Atmospheric Temperature, Deg. Fahr. Drum | Bead No. 1 | Bead No. 2 |
|---|---|---|---|---|---|
| *A*, with Drum No. 1 | 1 | 4.13 | 247 | 120 | 39 |
| | | 6.49 | 359 | 189 | .. |
| | | 8.07 | 458 | 251 | .. |
| *C*, with Drum No. 1 and Fan No. 1 | 2 | 4.50 | 224 | 98 | 36 |
| | | 6.50 | 320 | 148 | 39 |
| | | 8.63 | 391 | 186 | 41 |
| *C*, with Drum No. 1 and Fan No. 2 | 3 | 4.92 | 222 | 86 | .. |
| | | 9.05 | 381 | 154 | .. |
| *D*, with Drum No. 1 | 4 | 5.12 | 211 | 82 | 49 |
| | | 6.50 | 269 | 103 | 55 |
| | | 9.06 | 382 | 157 | 68 |
| | | 12.59 | 465 | 210 | 80 |
| *B*, withDrum No. 1 | 5 | 4.03 | 252 | 95 | 60 |
| | | 4.52 | 268 | 103 | 62 |
| | | 7.30 | 373 | 148 | 74 |
| | | 7.97 | 393 | 156 | 78 |
| | | 9.46 | 457 | 178 | 85 |
| *A*, with Drum No. 2 | 6 | 5.11 | 258 | 131 | 51 |
| | | 7.28 | 357 | 179 | 60 |
| | | 9.24 | 469 | 239 | 73 |
| *C*, with Drum No. 2 and Fan No. 1 | 7 | 5.52 | 262 | 105 | .. |
| | | 7.46 | 363 | 143 | .. |
| | | 10.23 | 476 | 197 | .. |
| *D*, with Drum No. 2 | 8 | 5.10 | 204 | 66 | 39 |
| | | 6.50 | 279 | 90 | 48 |
| | | 9.42 | 372 | 130 | 58 |
| | | 12.28 | 440 | 172 | 65 |

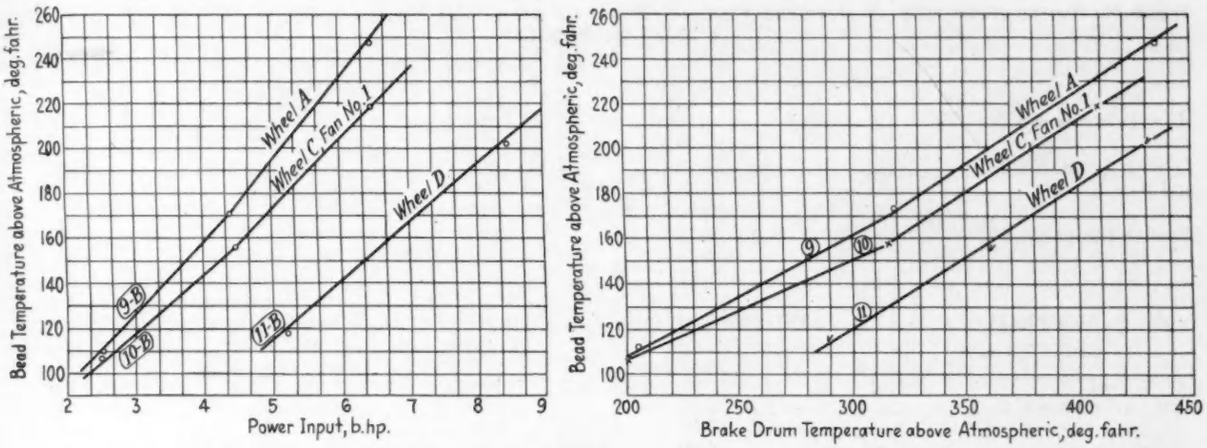


FIG. 9—COMPARISON OF HEATING OF THREE 20-IN. WHEELS IN TESTS BY BRAKE-HORSEPOWER-INPUT AND BY BRAKE-DRUM-TEMPERATURE METHODS AT A WHEEL SPEED EQUIVALENT TO A ROAD SPEED OF 15 M.P.H.

Both Charts Show the Wheels in the Same Relative Order but Curves at the Left, for Relation of Bead Temperature to Brake Horsepower, Show a Larger Differential Than Curves at the Right, for Relation of Bead Temperature to Brake-Drum Temperature. Note That for the Same Brake Horsepower the Bead Temperatures Are Higher at a Speed of 15 M.P.H. Than at 30 M.P.H., as Shown by Comparison with Fig. 1. Drum No. 1 Was Used in All Cases

TABLE 2—TEMPERATURE RISE AT A WHEEL SPEED OF 15 M.P.H. WITH DUAL TIRES LOADED TO 4400 LB. AND INFLATED TO 80-LB. PRESSURE

| Wheel | Curve, No. | Power Input, B.Hp. | Temperature Rise above Atmospheric Temperature, Deg. Fahr. Drum | Bead No. 1 | Bead No. 2 |
|---|---|---|---|---|---|
| *A*, with Drum No. 1 | 9 | 2.52 | 205 | 111 | 44 |
| | | 4.44 | 319 | 171 | 58 |
| | | 6.47 | 434 | 248 | 74 |
| *C*, with Drum No. 1 and Fan No. 1 | 10 | 2.52 | 200 | 106 | 46 |
| | | 4.50 | 310 | 157 | 48 |
| | | 6.48 | 409 | 218 | 58 |
| *D*, with Drum No. 1 | 11 | 5.23 | 289 | 115 | 56 |
| | | 6.48 | 361 | 155 | 70 |
| | | 8.48 | 431 | 203 | 87 |

in such a test sufficient added braking is used to nullify any difference there may be in two different wheels in this respect. This important consideration, together with the difference in spacing between the drum and rim of these two wheels, accounts for the change of order as shown in these two graphs. The data on the chart at the left in Fig. 8 take into consideration the relative cooling of the brake drums on the two wheels by both conduction and convection and are, without doubt, more nearly comparable with actual service conditions than the data given on the chart at the right.

### EFFECT OF SPEED VERY IMPORTANT

Comparison of the three 20-in. wheels at 15 m.p.h., as shown in Fig. 9, and at 10 m.p.h., as in Fig. 10, again shows these wheels in the same relative order but with

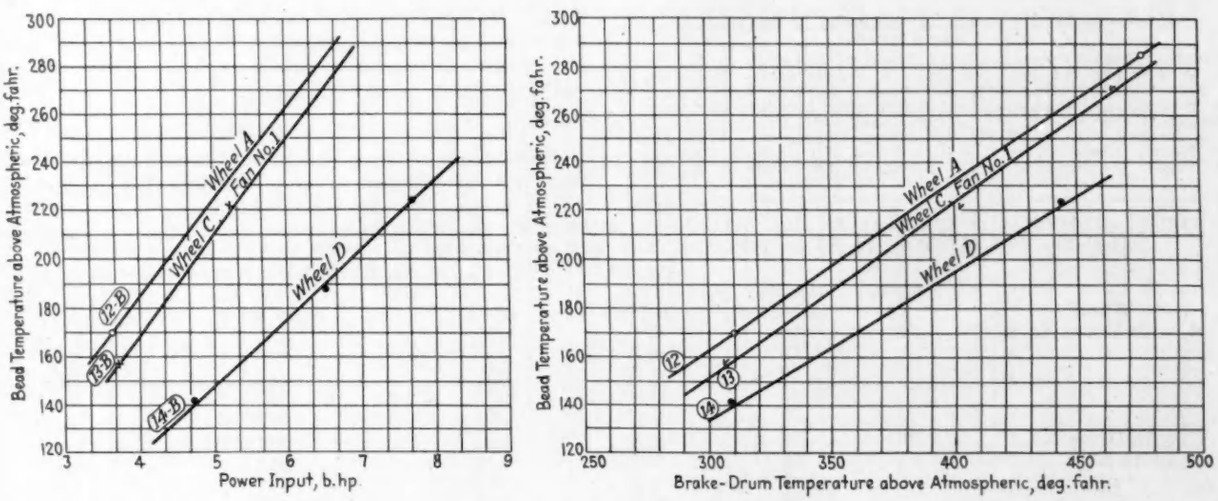


FIG. 10—EFFECT ON BEAD TEMPERATURE OF BRAKE-HORSEPOWER INPUT AND OF BRAKE-DRUM TEMPERATURE WITH 20-IN. WHEELS AT A WHEEL SPEED EQUIVALENT TO A ROAD SPEED OF 10 M.P.H.

Relative Order of the Wheels Is Unchanged but at the Lower Speed the Bead-Temperature Rise Is Greater in All Cases and the Relative Effect of the Fan on Wheel *C* Is Apparent. For the Same Brake Horsepower the Bead Temperatures for a Given Wheel Are Higher at 15 M.P.H. Than at 30 M.P.H., and the Temperature Rises More Rapidly at the Slower Speed as the Brake-Horsepower Input Is Increased as Will Be Seen by Comparison with Fig. 8

TABLE 3—TEMPERATURE RISE AT A WHEEL SPEED OF 10 M.P.H. WITH DUAL TIRES LOADED TO 4400 LB. AND INFLATED TO 80-LB. PRESSURE

| Wheel | Curve, No. | Power Input, B.Hp. | Temperature Rise above Atmospheric Temperature, Deg. Fahr. Drum | Bead No. 1 | Bead No. 2 |
|---|---|---|---|---|---|
| *A*, with Drum No. 1 | 12 | 3.62 | 310 | 169 | 55 |
| | | 6.57 | 475 | 285 | 91 |
| *C*, with Drum No. 1 and Fan No. 1 | 13 | 3.73 | 306 | 156 | .. |
| | | 5.23 | 402 | 221 | .. |
| | | 6.50 | 464 | 271 | .. |
| *D*, with Drum No. 1 | 14 | 4.73 | 309 | 140 | 64 |
| | | 6.50 | 400 | 188 | 73 |
| | | 7.77 | 443 | 223 | 100 |

TABLE 4—TEMPERATURE RISE AT A WHEEL SPEED OF 30 M.P.H. WITH NO LOAD AND TIRES INFLATED TO 80-LB. PRESSURE

| Wheel | Curve, No. | Power Input, B.Hp. | Temperature Rise above Atmospheric Temperature, Deg. Fahr. Drum | Bead No. 1 | Bead No. 2 |
|---|---|---|---|---|---|
| *A*, with Drum No. 1 | 15 | 4.75 | 271 | 129 | 33 |
| | | 6.60 | 356 | 183 | 47 |
| | | 8.60 | 479 | 255 | 74 |
| *C*, with Drum No. 1 and Fan No. 1 | 16 | 4.40 | 231 | 90 | .. |
| | | 6.60 | 325 | 140 | .. |
| | | 8.61 | 409 | 185 | .. |
| *D*, with Drum No. 1 | 17 | 4.10 | 172 | 48 | 22 |
| | | 6.60 | 272 | 82 | 36 |
| | | 9.20 | 365 | 127 | 50 |
| | | 12.20 | 461 | 187 | 65 |
| *B*, with Drum No. 1 | 18 | 4.94 | 267 | 70 | 26 |
| | | 6.83 | 364 | 102 | 33 |
| | | 9.07 | 458 | 134 | 62 |

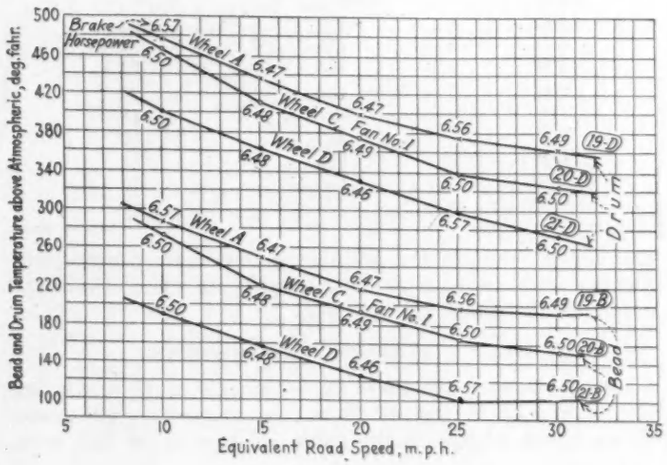


FIG. 11—EFFECT OF WHEEL SPEED ON RELATION OF BRAKE-DRUM AND BEAD TEMPERATURES

The Cooling Action Increases with Speed of Rotation in All Three 20-In. Wheels and the Relative Effect Is Substantially the Same in All. This Indicates that City Motorcoach Service with Frequent Stops Presents as Serious a Heat-Dissipating Problem as Intercity Service at Higher Speeds over Hilly Roads with Frequent Application of Brakes. In the Tests the Brake-Horsepower Was Maintained Nearly Constant at 6.50 Hp. Drum No. 1 Was Used with Each Wheel. Letter *D* after the Reference Numerals in the Rings Refers to Drum Temperature in Table 5

a larger differential when plotted on the basis of brake-horsepower. It is to be noted that for the same brake-horsepower at 15 m.p.h. the bead temperatures of a given wheel are higher than at 30 m.p.h. It will be noted further that the temperatures are still higher for a given brake-horsepower at a speed of 10 m.p.h.

The effect of speed on the relative cooling-action of the three 20-in. wheels, as shown by the data, was noted as of exceptional importance and it is believed that this is the first time that such data have been available. Accordingly, further tests were run at 20 and 25 m.p.h., using a constant brake-horsepower of 6.5. The complete data, under these conditions, are given in Fig. 11 and show that in all three wheels the cooling action increases with the speed of rotation between the limits tested, and that speed has substantially the same relative effect on each of the three wheels.

Intracity service on motorcoaches at low average speeds with frequent stops offers, therefore, as serious a problem on the dissipation of brake-drum heat as intercity service at higher speeds over roads of uneven contour with frequent use of brakes.

## TESTS OF HEAT-DISSIPATING DEVICES

A complete study of all known heat-dissipating devices was not attempted in the Sprague-dynamometer test.

TABLE 5—COMPARATIVE TEMPERATURE RISE WITH 20-IN. WHEELS AT VARIOUS SPEEDS, WITH DUAL TIRES, LOADED TO 4400 LB. AND INFLATED TO 80-LB. PRESSURE, AND ONE OF THE WHEELS FITTED WITH A COOLING FAN

| Wheel | Curve, No. | Speed, M.p.h. | Power Input, B.Hp. | Temperature Rise above Atmospheric Temperature, Deg. Fahr. Drum | Bead No. 1 | Bead No. 2 |
|---|---|---|---|---|---|---|
| *A*, with Drum No. 1 | 19 | 10 | 6.57 | 475 | 285 | 91 |
| | | 15 | 6.47 | 434 | 248 | 74 |
| | | 20 | 6.47 | 397 | 215 | 59 |
| | | 25 | 6.56 | 373 | 195 | 56 |
| | | 30 | 6.49 | 359 | 189 | .. |
| *C*, with Drum No. 1 and Fan No. 1 | 20 | 10 | 6.50 | 464 | 271 | .. |
| | | 15 | 6.48 | 409 | 218 | 58 |
| | | 20 | 6.49 | 374 | 191 | 49 |
| | | 25 | 6.50 | 334 | 162 | 43 |
| | | 30 | 6.50 | 320 | 148 | 39 |
| *D*, with Drum No. 1 | 21 | 10 | 6.50 | 400 | 188 | 73 |
| | | 15 | 6.48 | 361 | 155 | 70 |
| | | 20 | 6.46 | 328 | 125 | 60 |
| | | 25 | 6.57 | 295 | 101 | 50 |
| | | 30 | 6.50 | 269 | 103 | 55 |

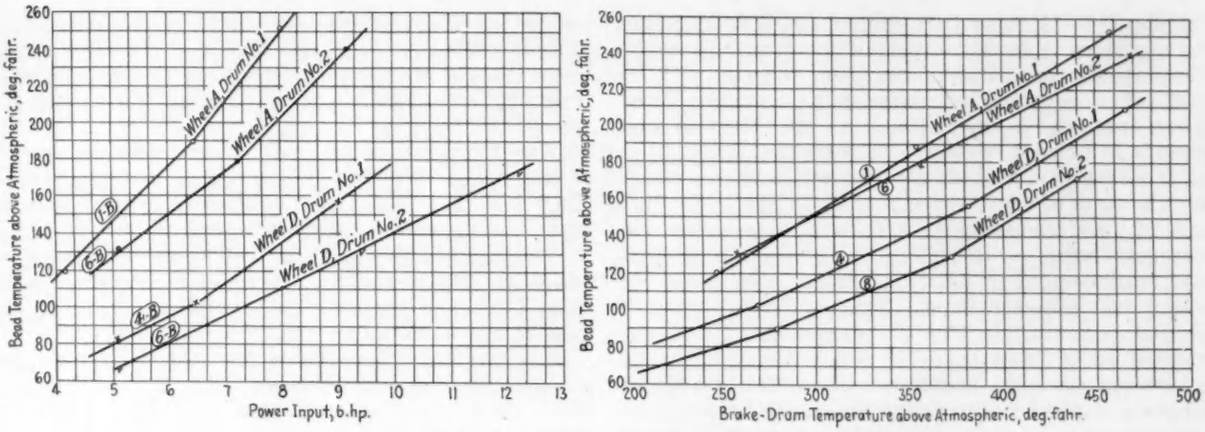


FIG. 12—HEAT-DISSIPATING EFFECT OF CHANGE IN BRAKE DRUM

Drum No. 2 Was Identical with Cast-Steel Drum No. 1 Except That (*a*) Radial Corrugations Extending from the Flange Part Way across the Outside Were Removed To Increase the Ventilating Space between the Drum and the Rim; (*b*) an Asbestos Covering Was Put On To Decrease Direct Radiation from the Drum to the Rim, and (*c*) the Band Which Closed the Space between the Dual Rims on Wheel *D* Was Added to Wheel *A* So That the Test Results Would Be Comparable. In Each Case Drum No. 2, Combined with the Deflector Band, Shows a Substantial Improvement in Heat Dissipation over Drum No. 1 without the Deflector Band. The Wheel Speed Was Equivalent to a Road Speed of 30 M.P.H. in Each Case. The Tests Shown Are Comparable with Those Shown in Fig. 13

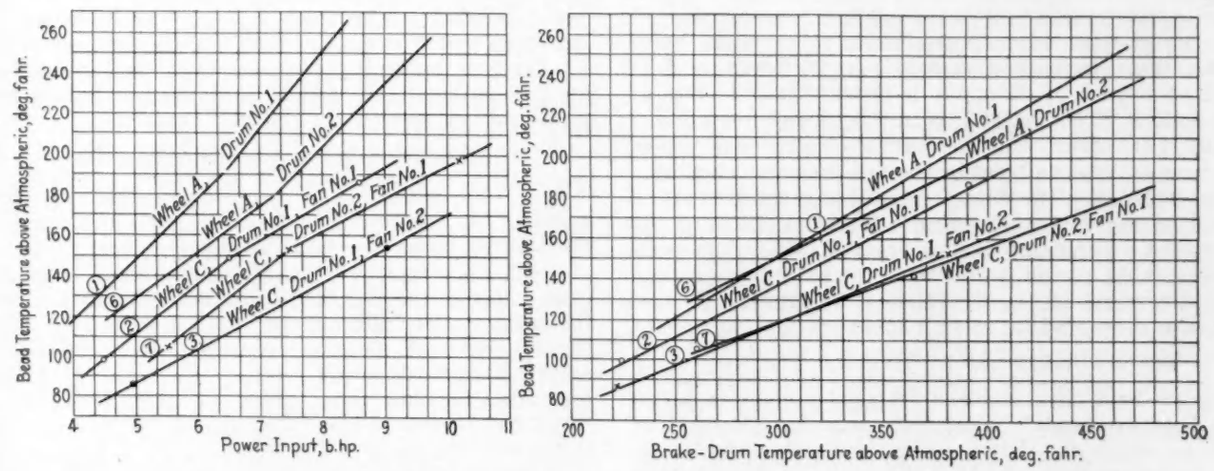
Bead Temperature above Atmospheric, deg. fahr.
Power Input, b.hp.
Wheel A, Drum No. 1
Wheel A, Drum No. 2
Wheel C, Drum No. 1, Fan No. 1
Wheel C, Drum No. 2, Fan No. 1
Wheel C, Drum No. 1, Fan No. 2
Brake-Drum Temperature above Atmospheric, deg. fahr.

FIG. 13—EFFECT OF FANS ON HEAT DISSIPATION WITH DRUMS NOS. 1 AND 2 AT A SPEED OF 30 M.P.H.

Fan No. 1 Was Encircled by a Band That Did Not Entirely Close the Space between the Dual Rims, Whereas the Band on Fan No. 2 Closed This Space and the Fan Vanes Were Set at a Different Angle Than in Fan No. 1 with the Object of Improving Ventilation. Each Had Eight Vanes. It Should Be Noted That in the Chart at the Right, for Relation of Bead Temperature to Drum Temperature, the Combination of Fan No. 2 with Drum No. 1 Produced Substantially the Same Cooling Effect as Fan No. 1 with the Improved Drum No. 2 on Wheel *C* as Shown in the Bottom Curves

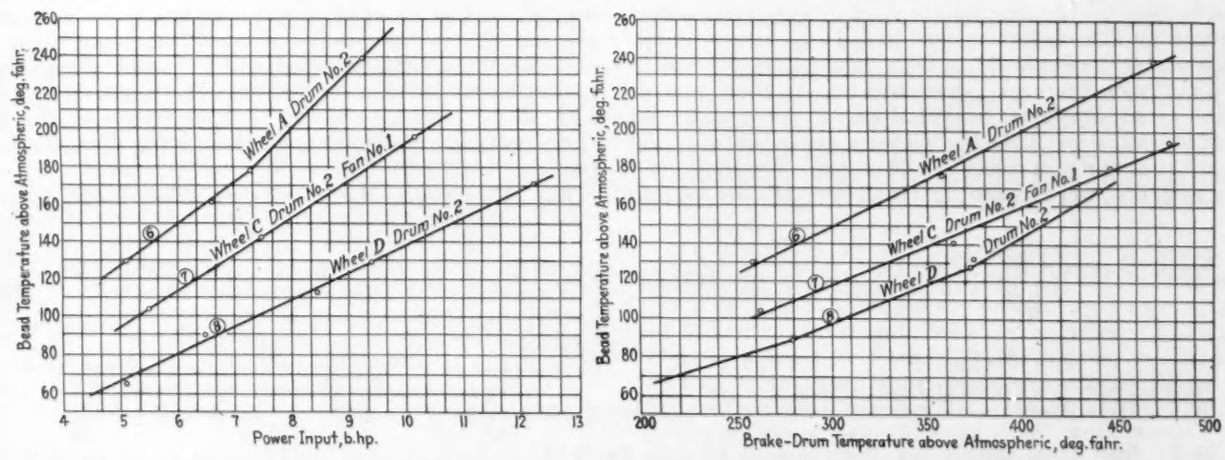
Bead Temperature above Atmospheric, deg. fahr.
Power Input, b.hp.
Wheel A, Drum No. 2
Wheel C, Drum No. 2, Fan No. 1
Wheel D, Drum No. 2
Brake-Drum Temperature above Atmospheric, deg. fahr.

FIG. 14—RISE OF BEAD TEMPERATURE WITH BRAKE DRUM THAT AFFORDED INCREASED VENTILATING SPACE BETWEEN DRUM AND WHEEL RIM AND THAT WAS COVERED WITH ASBESTOS

These Curves Are Comparable with Those in Fig. 8, Which Were Taken with the Wheels Fitted With Drum No. 1. The Wheel Speed in All Cases Was Equivalent to a Road Speed of 30 M.P.H.

The principal tests along this line consisted in removing the corrugations from the outside of the drum to increase the ventilating space between the drum and rim with the addition of a covering of asbestos to decrease direct radiation from the drum to the rim. At the same time the band which closed the space between the dual rims on wheel *D* was also added to the two other 20-in. wheels so that the tests of these three wheels, as shown in Figs. 12 and 13, are comparable. In each case drum No. 2, combined with a deflector band, showed a substantial improvement in heat dissipation over drum No. 1 without the deflector band. The results on wheel *C* are plotted separately in Fig. 13 because fan No. 1 was surrounded by a band that did not entirely close the space between the rims and also because fan No. 2, besides having a band fully closing the space between the rims, possessed vanes having a different angle designed to improve ventilation. It will be noted that fan No. 2 with drum No. 1 produced substantially the same cooling effect as the original fan No. 1 with the improved form of drum No. 2.

It is regrettable that the dynamometer set-up was dismantled before the combined effect of fan No. 2 with drum No. 2 had been tested on wheel *C*. Such a combination probably would have resulted in a still further lowering of bead temperatures for a given brake horsepower. For purposes of comparison, the effect of drum No. 2 at a speed of 30 m.p.h. is shown again in Fig. 14, thus making it comparable with Fig. 8 and drum No. 1.

Only one other heat-dissipating device was tested. This was an external ventilating fan, motor-driven, playing on wheel *A*. The data for this are given in Table 6. It will be noted that the most efficient position of this fan was at an angle of 45 deg. from the line of the axle, whereas, when placed in line with the axle, there seemed to be a reversal of the direction of ventilation through the brake shoes and its cooling effect was not so marked.

### Beads Unaffected by Heat of Flexion

Reports have been received frequently of tires failing from "burned beads" on long-distance hauls over level roads on which the driver was positive that the brakes were used very little and that the brakes were not dragging. Assertions were made that these tires failed due to heat generated by flexing of the tire and not to brake heat. The question of the effect of heat from tire flexing was, therefore, investigated carefully with the Sprague dynamometer. The major distortions in a tire are produced by load and by traction torque. The heating effect produced by load deflection was tested by placing on the dynamometer a single 32 x 6-in. heavy-duty tire on a 20-in. rim with the brake shoes removed so there was no possibility of dragging brakes, loading the tire to the standard rating of 2200 lb. with an air-inflation of 80 lb., and running at a speed of 30 m.p.h. Thermocouple measurements were made by insertion of the couple under the tread of the tire and by a permanent thermocouple placed between the bead and the rim. This test was made with the generator unit of the dynamometer running free, equivalent to running the tire on a level road.

The temperature at equilibrium under the tread was found to be 227.0 deg. fahr., the room temperature was 84.0 deg. fahr. and the rise above atmospheric temperature was therefore 143.0 deg. fahr.

The bead-temperature of the same tire became constant at 106.5 deg. fahr., the room temperature was 80.0 deg. fahr. and the rise above atmospheric temperature was therefore 26.5 deg. fahr.

The heating effect from the deflection produced by traction torque was checked with a second set-up on both 34 x 7.50-in. balloon tires and 32 x 6.00-in. high-pressure truck-tires, using a 12-hp. drag on the generator or road-wheel unit, which was equivalent to a grade of 4.4 per cent. Temperatures at the bead became constant at the same temperature irrespective of whether the tire was running "on the level" or "up-grade."

It can, therefore, be stated with assurance that a tire which is properly loaded and inflated will not generate sufficient heat to cause bead failures. Over-load or under-inflation undoubtedly will produce a greater heating effect which, combined with drastic mechanical action on the tire, will result in premature failure, but even such an extreme case will not produce the failure known and easily recognized as "burned beads."

### Bead Burning Due to External Heat

In a tire unaffected by brake-drum heat, the hottest point in the tire is found to be in the outer portion of the carcass under the tread at the shoulder. The travel of heat, therefore, will be from the tread toward the bead, through the sidewalls, tube and internal air, chiefly the last. A tire failure from "heat produced by flexing" should, therefore, show the greatest effect of heat in the outer portions of the tire. Every tire, however, to which our attention has been called as having failed from "heat due to tire flexing" has shown the greatest effect of heat at the bead, which is prima facie evidence that the failure was due, not to tire heat, but to some external source of heat.

In tires subjected to heat from brake drums, a reversal of normal conditions occurs and the flow of heat is from the bead to the tread. It is evident, therefore, that heat from tire flexing will have a certain influence on the dissipation of heat from the beads by decreasing the temperature gradient between bead and tread and will maintain thereby a somewhat higher temperature at the bead. To show this effect, the tests recorded on Fig. 15 were repeated at no load by removing the tires from the road wheel, which was allowed to remain stationary.

Fig. 15 records the results of these tests, together

TABLE 6—EFFECT OF COOLING FAN[3] ON WHEEL *A* AT A WHEEL SPEED OF 30 M.P.H., WITH DUAL TIRES, LOADED TO 4400 LB. AND INFLATED TO 80-LB. PRESSURE, AND A BRAKE-HORSEPOWER INPUT OF 6.49 HP.

| Fan Position | Temperature Rise above Atmospheric Temperature, Deg. Fahr. Drum | Bead No. 1 | Bead No. 2 |
|---|---|---|---|
| No Fan. See Top Curve at Right in Fig. 8 | 359 | 189 | .. |
| 30 In. from Tires and Blowing Normal to Axle, Directly between Tires | 283 | 135 | 48 |
| 30 In. from Tires and 45 Deg. from Line of Axle, in Front of Drum | 261 | 116 | 44 |
| 30 In. from Tires in Line with Axle, in Front of Drum | 266 | 120 | 44 |

[3] Robbins & Myers; speed, 1150 r.p.m., 60 cycles, 110 volts. Air velocity at 30 in. from fan, 800 ft. per min., determined by Keuffel & Esser anemometer.

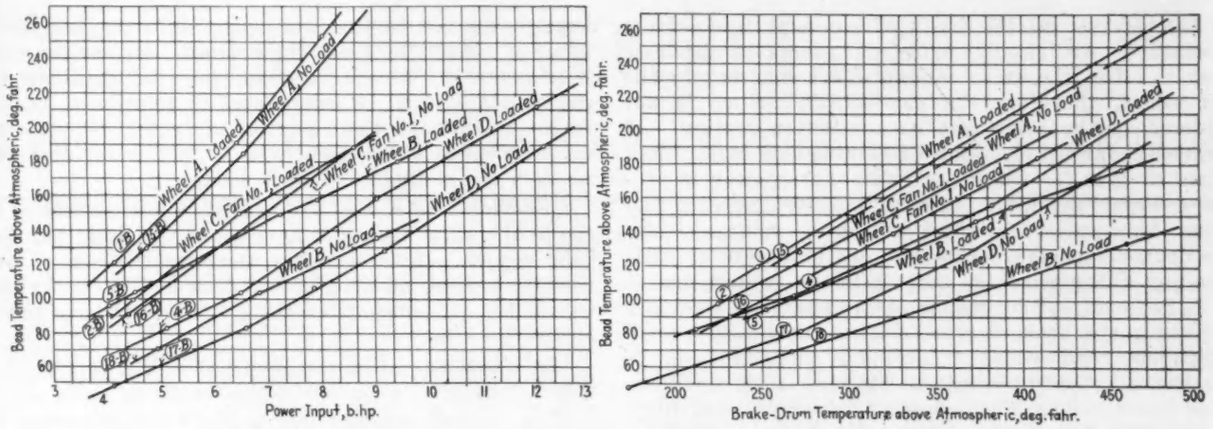
Bead Temperature above Atmospheric, deg. fahr.
Power Input, b.hp.
Wheel A, Loaded
Wheel A, No Load
Wheel C, Fan No. 1, Loaded
Wheel C, Fan No. 1, No Load
Wheel B, Loaded
Wheel D, Loaded
Wheel D, No Load
Wheel B, No Load
Brake-Drum Temperature above Atmospheric, deg. fahr.

FIG. 15—EFFECT OF HEAT FROM TIRE FLEXING ON RELATION OF DRUM TEMPERATURE TO BEAD TEMPERATURE

In One Series of Tests the Wheels and Tires Were Run with 80-Lb. Air-Inflation, but without Load, and for Comparison the Curves for the Same Equipment with Standard Load of 4400 Lb. and the Same Speed of 30 M.P.H. Are Redrawn on the Same Charts. In Every Case More Rapid Dissipation of Heat from the Bead through the Tire Was Made Possible by Increase of the Temperature Gradient Due to Removal of the Effect Caused by Tire Flexing. This Difference in Dissipation Was Comparatively Slight and the Close Relation of the Curves for a Given Wheel Indicates That Heat from Tire Flexing Is Not an Important Factor in Bead Failure, Although Excessive Flexing Accentuates the Effect of Brake-Drum Heat

with the original loaded tests, regraphed for comparison. In every test the removal of the effect of heat due to tire flexing increased the temperature gradient between the bead and tread, and therefore permitted more rapid dissipation of heat from the bead through the tire itself. The difference, however, is comparatively slight and again the conclusion can be drawn that heat from tire flexing is not an important factor in the failure of tires from brake-drum heat.

## SUMMARY AND CONCLUSIONS

(1) The relative cooling-action of the four wheels tested is found to be substantially in the same order as reported in the Goodyear and Budd tests, except that wheel *C* with drum No. 1 and fan No. 2 was found to be comparable with wheel *D*

(2) Comparison of the four wheels on the basis of constant brake-horsepower shows a wider difference, in the same order, than comparison on the basis of constant brake-drum temperature, as was prophesied in the Goodyear report. Comparison on the basis of constant brake-horsepower more nearly reflects the relative value of the wheels in actual service

(3) It appears that the greater the cooling action of a given wheel is, the greater is the error in comparing this with other wheels on the basis of constant brake-drum temperature, and the greater is the necessity of a comparison on the basis of constant brake-horsepower

(4) For the heat-dissipating devices tested, substantially the same relative improvement was noted on all wheels

(5) The cooling properties of a wheel increase with the speed of rotation for a given brake-horsepower input

(6) Variations in speed did not materially change the relative cooling-action of the three 20-in. wheels

(7) Forced ventilation by external fans is indicated as an aid for the dissipation of brake-drum heat. The slower the speed of a motorcoach is the greater should be the effect of forced ventilation

(8) The tire trouble known as "burned beads" cannot result from "heat due to tire flexing" but must be attributed to brake-drum heat caused by dragging brakes or excessive use of brakes

(9) Excessive flexing in a tire due to under-inflation, heavy loading or high speed will accentuate the effect of brake-drum heat on the bead by decreasing the temperature gradient between the bead and tread

## THE DISCUSSION

QUESTION:—Do you recommend the use of asbestos between the drum and rim, and, if so, is not this passing the problem on to the brake and axle manufacturers, since the heat generated by braking must be dissipated in some way?

C. W. BEDFORD:—I believe that the closer the rim is to the drum, the greater is the necessity for preventing radiation from the brake. While it is improper to lag a heat-radiating device, a compromise may be desirable in some circumstances.

QUESTION:—Why not line the inside of the wheel rim with asbestos instead of wrapping the drum?

MR. BEDFORD:—The suggestion is of interest. The asbestos next to the rim would allow convection cooling of the drum and at the same time intercept radiation from the brake.

QUESTION:—Would not the use of brakes on every available wheel reduce the quantity of heat to be dissipated by each wheel and thus reduce the amount of heat absorbed by each tire?

MR. BEDFORD:—That is one of the advantages claimed by the four-wheel-brake engineers. I have heard or read a statement that with brakes on four wheels the energy required on a given drum is reduced about 40 per cent.

QUESTION:—Fig. 11 seems to show that ventilation operates wholly on the brake drum and not on the bead or rim. Is this correct?

MR. BEDFORD:—That is not the case. However, considerations which show the advantages of testing on the basis of brake horsepower instead of constant brake-drum temperature apply only to the drum. The heat is dissipated from the drum by three means: convection, conduction and radiation. It is also dissipated from the rim in the same way but only the cooling of the drum, as the source of heat, is considered.

If of two different heat-dissipating devices one has greater heat-conduction through metal parts from the drum to the spokes, hub and axle, that device should run the cooler, so far as bead temperature is concerned; but

if a test is run on the basis of constant brake-drum temperature, we supply sufficient increased braking energy to offset the extra cooling effect inherent in the cooler wheel, therefore the conduction cooling is not considered in the same way as convection cooling. If one wheel is fanned and cooled by convection better than another so that for the given brake horsepower it produces a lower drum-temperature but we increase the braking horsepower for that set-up to maintain the lower drum-temperature constant, we eliminate or neutralize any difference there may be in the two set-ups.

A. L. SCHOFF[4]:—Why was the test apparatus arranged to deliver to the brake-drum the same brake-horsepower for different road speeds? Inasmuch as horsepower is a function of speed, it seems to me that, to make these heating tests comparable, constant torque should have been maintained regardless of the speed, rather than to maintain constant brake-horsepower, which with decreasing speeds means increased torque. This would, of course, mean a dissipation of less heat at the slower speeds than was the case under the conditions of the test as run, and, while it still might show an increase in brake-drum temperatures at slow speeds over high speeds, I do not believe the increase would be as high by any means as shown in the results presented in the paper.

Another question is this: Inasmuch as in practically all curves shown the heat was considerably greater on the disc wheel, in which the ventilation is poorer than on the two other types of wheel, are any data available to show what the relative temperatures would be on a wood-spoke wheel which not only would provide good ventilation but would prevent any direct conduction of heat from the brake-drum through the wheel to the rim?

MR. BEDFORD:—In motorcoach operation neither brake-horsepower nor torque is a constant factor and the element of time is of great importance. Either the brake-horsepower or torque relations may be obtained from the data and must be interpreted in relation to the time factor in applying them to road surfaces. No tests were made by us on wood-spoke wheels and, so far as I know, no data are available on such wheels.

QUESTION:—Mr. Bedford stated in his summary that the final results are equal to those of the Budd and Goodyear report. If we compare steel-spoke wheel *C*, using the No. 1 drum and No. 2 fan, does not its performance equal that of steel-spoke wheel *D*, which has the demountable rim?

MR. BEDFORD:—Fig. 15 shows that the curves of the wheels referred to lie very close together.

E. A. CLARK[5]:—It is noted in Mr. Bedford's paper that the condition to which he refers is one which existed in 1925. Since that time, the entire season of 1926 went by and we are now into 1927, and it is significant to recall that in neither 1926 nor so far this year has the question of wheel-rim heating been up for consideration. In May, 1926, a great deal of valuable time was expended by representatives of tire companies, wheel companies and others interested, in a prolonged test in Kansas City under the auspices of the Kansas City Railway Co. The final result from more than 80 tests, no two of which were comparable, was that 90 per cent of the trouble lay in the improper adjustment of brakes to prevent dragging.

The relief that has come to the occasional cases of actual heat-trouble that did exist has arrived through the proper correlating by motorcoach manufacturers of brake-drum and tire rim as to clearances. Efforts to eliminate heat by fans or other methods of this description lead to such an impractical solution from an actual operating standpoint as to relegate them to a purely theoretical consideration.

[4] M.S.A.E.—Assistant vice-president, Overman Cushion Tire Co., Inc., New York City.
[5] Budd Wheel Co., Philadelphia.

# MOTOR-VEHICLE SPEED IN CONNECTICUT

THE existing Connecticut law regarding speed of motor-vehicles on the highways is based upon the theory of reasonableness. Whatever speed is reasonable for all the conditions existing at the place in question is proper. But the law includes certain prima-facie speeds. As established, the most important are 20 m.p.h. in thickly settled portions of the State and 30 m.p.h. on the open highway. To bring out more clearly what a prima-facie speed means, it may be explained that the effect of it is to shift the burden of proof and has nothing whatever to do with the principle which remains to govern all cases by measurement as to whether or not it is reasonable. These prima-facie speeds do not change this law but provide a method for its application. If, where the prima-facie speed is 30 m.p.h., an operator is going 40 m.p.h. and an officer arrests him for speeding, when the officer proves that the arrested person was actually exceeding 30 m.p.h., the burden of proof shifts and the arrested person is required to prove that the speed at which he was going was reasonable considering all the conditions. If he can so prove, he will be acquitted. The matter of a change in this law came before the Legislative Committee on Motor-Vehicles in a bill to increase these prima-facie speeds. But the hearings developed a fundamental objection to any prima-facie speed at all on the ground that to have such creates a situation where a person may be obliged to prove his innocence. This is contrary to the fundamental tenets of criminal law, to custom and to the spirit of self-government. Therefore, it is proposed that all prima-facie speeds be taken out and that the speed law be left without any indication as to miles per hour, so that, should the proposed bill become law, there could be no arrests except on the ground that a speed was unreasonable. The arresting officer will then in every case have to prove the unreasonableness of the speed to secure a conviction.

This is a great step toward justice, for the improved construction of cars and the betterment in highways have created conditions under which the old prima-facie speeds are commonly exceeded by everyone. It is a step in advance from the standpoint of putting more responsibility upon the operator of a car. It should cause the exercise of conservative judgment in driving. There is apt to be too much of a tendency to direct operators. The more traffic direction there is, the less initiative and responsibility an operator can be expected to exercise. Direction is necessary. It does not now exist in many places where it should. But it will not do to *over-direct*. Responsibility and proper exercise of judgment and care must be preserved. They will make for high-mindedness and self-reliance, both of which are essential in good operation.—From Bulletin No. 41 of the State of Connecticut Department of Motor Vehicles.

# Internal Wheel-Brakes

## Discussion of H. D. Church's Semi-Annual Meeting Paper[1]

ABSTRACT

The paper deals primarily with internal wheel-brakes for trucks and motorcoaches, but passenger-car brakes with similar characteristics are considered possible. A simple two-shoe internal-expanding type developed mainly by empirical methods is found to be the most practical solution in spite of relatively low circumferential contact. Self-energization is necessary to reduce driver effort with normal pedal-travel. The factors controlling self-energization are explained in detail, and the effect of difference in the coefficient of friction of brake-linings is noted.

Distortion of brake-drum and brake-shoes must be limited by a drum of heavy section and by extremely rigid shoes. Rotation of cam with respect to self-energizing shoe should tend to deflect the toe of shoe away from brake-drum surface. A floating cam is necessary to balance unequal wear on the brake-shoes and assure adequate braking with normal pedal-pressure.

High-grade cast-iron brake-drums have been found to be generally more effective than steel drums with fabric-lined brakes.

Air-brakes of the same general design, with metal-to-metal surfaces, are used in all four wheels of the six-cylinder motorcoach, but the floating-cam construction is unnecessary in this case. The metal-to-metal combination of high-carbon drums with low-carbon shoes is found to be effective under all weather conditions, and is regarded as the solution of the problem of brakes with adequate life and effective heat-dissipation for heavy, high-speed motorcoaches.

Various problems of metal-brake design were brought to attention in the discussion, but most interest was shown in the questions of the relative cost of brakes of different types and of heat-dissipating qualities.

H. D. HUKILL[2]:—This excellent paper of Mr. Church's deals with a problem whose successful solution is essential to the development of the high-speed heavy-duty automotive vehicle of today. It is interesting to note that the various points Mr. Church mentions as having been proved during experimental road-work have been borne out by our own test-work and observance of equipment in service.

It has been found most important to use rigid supports of ample cross-section for the anchor pins and camshafts, and it is also very important that anchor pins and camshaft bearings be correctly and accurately located. The brake-drum surface must be perfectly concentric with the hub bore, and where drum and hub are a two-piece unit the drum surface should be finished after the hub and drum are assembled, centering from the hub bore mounted on a suitable mandrel.

To assure smooth brake-action the rotation of the cam during brake application should be in the same direction as that of the brake-drum. This was mentioned by Mr. Church and the reason for it is apparent from his Fig. 1.

The heat-dissipating properties of the all-metal brake were thoroughly demonstrated in a series of tests made in the Allegheny Mountains last August, using a Model 50-B four-cylinder 29-passenger White motorcoach with standard factory brake-equipment of air-operated all-metal brakes on the rear wheels only. Test tire-equipment was Goodyear 36 x 8.25-in. dual balloons on 20-in. rims. This coach, loaded to 17,000 lb., was coasted with gears in neutral at a constant speed of 15 m.p.h. down an approximately uniform 8¼-per cent grade 14,000 ft. long. The speed was controlled entirely by the rear-wheel metal brakes.

The maximum brake-drum temperature encountered on any run, measured by calibrated thermocouples located at the center of the brake-drum face and ⅛ in. below the friction surface, read 30 sec. after stopping at the foot of the grade, was 625 deg. fahr. The maximum tire-temperature, as determined with thermocouples located between the tire bead and the metal tire-rim on the inner tire, was 145 deg. fahr. This was found 15 or 20 min. after ending the run, when the heat from the brakes had had an opportunity to be conducted into the adjacent parts without the influence of any cooling drafts such as come into play during actual operation. The maximum hub-temperature, determined in the same way, was 145 deg. fahr., and the maximum axle-housing temperature, taken at a point under the spring pad, was slightly lower, being 125 deg. fahr.

Another interesting point is that the brake-shoes or heads, taking a slightly higher temperature than the brake-drum, expanded at approximately the same rate as the brake-drum; hence there is no "run-out" of the brake at any time, brake-chamber travel remaining constant under all conditions.

A. Y. DODGE[3]:—It would be interesting to know more concerning the specifications of the material in the shoes and the drum of the metal brake that is under discussion.

F. G. ALBORN[4]:—The carbon content of the shoes is between 23 and 28 point and that of the drum is about 50.

W. J. BAUMGARTNER[5]:—What brings the floating cam-mechanism back into position after it has been jarred out of place by road shocks?

MR. ALBORN:—We have encountered no condition of that kind. There is no indication of any drag due to the shoe falling.

CHAIRMAN G. A. GREEN[6]:—Why do you use cast iron for the shoes of the small vehicle and cast steel for the big one?

MR. ALBORN:—We have found that steel is better for metal brakes, but because the friction-fabric lining tends to pick off metal particles from a steel casting we use cast iron, which shows no such tendency, for our fabric-

[1] The paper with illustrations was printed in the June, 1927, issue of THE JOURNAL. The author is director of engineering, White Motor Co., Cleveland.

[2] M.S.A.E.—Westinghouse Air Brake Co., Wilmerding, Pa.

[3] M.S.A.E.—Chief engineer, Bendix Brake Corporation, South Bend, Ind.

[4] M.S.A.E.—Assistant chief engineer, White Motor Co., Cleveland.

[5] M.S.A.E.—Chief engineer, Garford Truck Co., Lima, Ohio.

[6] M.S.A.E.—Vice-president in charge of engineering, Yellow Truck & Coach Mfg. Co., Pontiac, Mich.

lined brakes. The cast iron we use has a very high combined-carbon content, with the graphite carbon present in the shape of small flakes peculiarly interlacing. It is very close grained.

CHAIRMAN GREEN:—Are we to assume that you use cast-iron brake-drums with the fabric-lined shoes and cast-steel drums with the metal throw-away-type shoes?

MR. ALBORN:—That is right.

### THE QUESTION OF LIFE

CHAIRMAN GREEN:—Can you give us some idea of the mileage that may be expected from the throw-away shoes? In the early part of your paper you mentioned tests where brake applications averaged one for each 0.9 mile. Were you referring to metal brakes or to fabric-lined brakes?

MR. ALBORN:—We were referring then to the lined brakes.

CHAIRMAN GREEN:—Can you give us some figures as to the possible mileage of metal throw-away-type shoes?

MR. ALBORN:—We are getting about the same mileage with the metal-to-metal throw-away type of shoe as with the lined type.

CHAIRMAN GREEN:—Then for approximately one stop per mile you can expect 25,000 miles from the shoes?

MR. ALBORN:—That is correct.

CHAIRMAN GREEN:—Suppose you make five stops per mile, would the life of the shoes be reduced to 5000 miles?

MR. ALBORN:—With an increased number of stops, the speeds at which braking is done will be lower, and as the braking effort required to accomplish a stop is proportional to the square of the velocity, under the conditions Mr. Green mentions we would get approximately the same life of the brake-shoes as under the conditions mentioned in the paper.

CHAIRMAN GREEN:—We cannot control that, can we? We must expect as violent an application as the operator can make. What will the throw-away shoes cost per mile in operations where they are called upon for five stops per mile?

MR. ALBORN:—My opinion is that the cost would not be any greater than the cost of lining.

CHAIRMAN GREEN:—What will be the ratio of wear for drums and shoes? Can we expect a drum to out-wear two or more sets of shoes?

MR. ALBORN:—We are finding about three sets of shoes are required to one drum. On tests which we have run we have not worn out a brake-drum. In the first 25,000 miles we had approximately 1/16-in. wear on the drum diameter. The brake-drums can stand as much as ¼-in. wear before it is necessary to replace them.

### DIFFICULTY WITH LINERS

A. J. SCAIFE[7]:—We have found that in using liners in the drum and in the shoe it is practically impossible to get a perfect fit between the liner of the shoe and the liner of the drum. The resulting air-gap between the drum and the liner acts as insulation against heat radiation. If the brake-drum liner is not perfectly true it may cause a slight buckling or distortion unless the liners are very carefully run in at the start. The life of the brake can be greatly increased or shortened, according to the methods used for the run-in. Also, we have found that the heat radiated to the drums, as brought out in the paper, will cause drum distortion. The distortion is very slight and occurs only during the life of the first set of liners. If new liners are put in without truing the drum, serious difficulty may result from a bad out-of-round condition. We have found the second set of liners to be eccentric from 0.004 to 0.015 in.

When it is necessary to remove the drum and take it to a machine-shop to be trued, the labor expense plus the parts is in many cases greater than the cost of a solid drum, especially with companies not having their own machine-shops. With some of the larger transportation companies it is not so serious because they have their own shops with machinery of sufficient size to do this work.

CHAIRMAN GREEN:—Coming back to the question of cost, which is surely a controlling issue, and assuming a case in which there are five stops per mile, suppose that the shoes will cost the operator about $50 per set and last for 5000 miles, and that the drums will cost $100 per set and require renewing after 15,000 miles, then the cost per mile will be 1 cent for shoes and 2/3 cent for drums; total, 1 2/3 cents. Of course this is an extreme case; nevertheless there are plenty of operations with more than five stops per mile, and it is inconceivable that any operator can afford to spend 1 2/3 cents per mile for shoe and drum renewals. Such an expenditure should go a long way toward the complete vehicle maintenance.

Assuming that the brake-drums are made from the proper material and that fabric-lined brake-shoes are used, the drum life is almost unlimited. Take, for example, the Chicago Motor Coach Co., operating approximately 500 vehicles, all of which have averaged over 100,000 miles. In no case has there been a brake-drum replacement since the vehicles were first put into service. These drums are made from hot-pressed, heat-treated, high-carbon steel. There are some definite advantages in the use of metal-to-metal brakes, yet we cannot eliminate the question of cost.

### EFFECT OF HEAT ON TIRES

MR. ALBORN:—Is it really a question of cost? With the heavy, high-speed motorcoach the question of heat is to be considered. Is it not cheaper to save tires at the expense of brake-shoes? With the metal-to-metal brake the shoes are more completely available for heat radiation to the air than is the case with the friction-material-lined shoe. We have demonstrated by repeated tests that the metal-to-metal brake absorbs not only several times the braking energy before the brake-drum reaches its maximum temperature, but that the maximum temperature will be materially lower than with lined brakes. This difference may mean the difference between saving and ruining the tires. There is the additional feature that Mr. Church pointed out in his paper, that the metal-to-metal brake is the safest brake we know of today, as its action is very little affected by oil, water or grit.

CHAIRMAN GREEN:—Must we infer that your choice of 20-in. rims has something to do with your liking for the metal-to-metal brake?

MR. ALBORN:—There is no doubt that it has. We need low floors in modern motorcoaches, and with any rim larger than 20 in. the tire needed is of such size that it makes the wheelhouse project into the body too much.

B. J. LEMON[8]:—Were any heat-tests between the bead of the tire and the rim made with brakes other than the metal-to-metal which were reported as showing a tem-

[7] M.S.A.E.—Field engineer, White Motor Co., Cleveland.

[8] M.S.A.E.—Automotive contact engineer, United States Rubber Co., Detroit.

perature of 145 deg. fahr. 15 or 20 min. after the downhill run of 2½ miles was completed?

C. A. OHL[9]:—This is the only report I have.

### FITTING SHOES IN WORN DRUMS

R. S. SANFORD[10]:—What is the operation of the metal-to-metal brake like after a drum is worn and the shoes are replaced by new ones? The diameter of the drum will then be greater than 17¼ in. and the shoe will not fit. Is not the operation of the brake erratic until the shoes wear to fit the drum?

MR. ALBORN:—When the first replacement of shoes is necessary, with a wear of approximately 1/16 in. on the drum diameter, it is customary to run-in the new shoes under full load with grease on the surfaces. This overcomes any irregularity in brake action and the grease prevents scoring.

TORE FRANZEN[11]:—By using cast-iron drums the unsprung weight is increased, is it not?

MR. ALBORN:—There is no necessity of that. The cast iron we are using has a tensile-strength of 39,000 lb. per sq. in., and on all our truck models we have replaced cast-steel drums with cast-iron drums without an increase in section.

CHAIRMAN GREEN:—Must we assume that brake-linings are a thing of the past? Is there not some brake-lining manufacturer here who will say something in favor of the brake-lining that we have used for so long?

MR. ALBORN:—It was pointed out that the metal-to-metal-lined brake was found necessary on the high-speed, heavy-weight motorcoach, but we are using the fabric-lined brake elsewhere.

MR. DODGE:—I believe the paper indicates that two rear brakes are used with fabric lining on a 21-passenger motorcoach which travels up to 40 m.p.h. Surely there is not so much difference between 40 and 60 m.p.h. as between two and four brakes or between two brakes with fabric lining and four metal-to-metal brakes.

MR. ALBORN:—The six-cylinder high-powered motorcoach will serve more generally in long runs and under conditions where metal-to-metal brakes are necessary because of the heat generated. Motorcoaches of this size may be fitted with bodies that will permit a greater percentage of overloading than with the smaller vehicle, on which we have definitely limited the body capacity.

MR. SANFORD:—Has Mr. Alborn's company made comparative tests of air-operated systems with fabric linings and with metal-to-metal surfaces? Is it necessary to use metal-to-metal brakes with air operation, or can satisfactory performance be had with fabric-lined brakes?

### METAL SURFACES REQUIRE POWER APPLICATION

MR. ALBORN:—We are using metal-to-metal brakes on our large motorcoach because of the heat conditions. Air is required for the application of our metal-to-metal brakes.

MR. SCAIFE:—The cost of the air-operated system is several hundred dollars more than for the direct-operated system and this greater cost is included in the first cost of the vehicle. As Mr. Alborn stated, it is necessary to use air with metal-to-metal shoes to get sufficiently effective braking. Fabric lining has been tried, and it cannot be so successful as the metal-to-metal surfaces.

MR. BAUMGARTNER:—We have used both fabric and metal lining on four-wheel air-brakes and have found that, if the grease can be kept off the lining, the fabric-lined brake is the better.

A. M. YOCOM[12]:—Six months ago many manufacturers were experimenting with metal brake-linings and were not willing to test the newer asbestos-fiber linings. This attitude now seems to have changed, as these same manufacturers are again testing and using asbestos-fiber linings. One motorcoach fleet-owner says that metal linings cost 5 times as much as asbestos-fiber linings and never last more than 1½ times as long; besides this there were grabbing, noise, scoring, and other brake ills to a greater extent than with the fabric linings. A metal drum and metal lining combination may be possible that will give good results, but it seems that metal woven or imbedded in asbestos fiber will give the best results, as there is enough resiliency in such a mixture to give the proper conformity to the drum. Because of expansion from brake heat and deflection from brake loads, there are variations in the lining contact. A slight amount of resilience in the lining compensates for these variations. Added to the conforming qualities of asbestos is its ability to maintain more friction when it becomes greasy than a metal lining would have under identical conditions.

### METAL-TO-METAL BRAKE DIFFICULTIES

M. C. HORINE[13]:—In our experiments with metal-to-metal brakes we have had two little difficulties upon which Mr. Church and Mr. Alborn can perhaps throw light. One is that the great heat of the metal-to-metal brake softens the brake-shoe spring. It has been found necessary to use much heavier springs than with fabric-lined brakes. Another result of this brake-shoe heat is difficulty in the lubrication of the hinge pins and the brake camshaft.

MR. ALBORN:—We have experienced no trouble with the retracting springs on our metal-to-metal brakes. The springs attach to light ears on the brake-shoe and the hooks at each end are separate from the main body of the spring, so the heat transfer is very slow. We have had no difficulty with the lubrication because, with air, we have a means of application that makes the brake effective if the lubrication of these points is neglected.

MR. HORINE:—The difficulty is that the brakes drag, not because the spring stretches, but because of faulty lubrication after the brakes have been heated.

CHAIRMAN GREEN:—Our experiments confirm some of the points that have been brought up in favor of metal-to-metal shoes, but we always come back to the question of cost. Can we afford to use throw-away-type shoes? Why should we not line the steel shoes and thus save the major part of the cost?

MR. SCAIFE:—Mr. Fageol is using this system on his new motorcoach, and possibly it has many advantages. Mr. Alborn has pointed out that our tests so far have been satisfactory with both. There is nothing to prevent using lined shoes without any other change in construction. Inasmuch as the shoe does not rotate, high points on the shoe do not make so much difference as high points on the drum. We believe that the air-brake is best for a heavy motorcoach, and it is an open question whether fitting new liners is cheaper than putting in new shoes.

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[9] A.S.A.E.—Automotive representative, Westinghouse Air Brake Co., Chicago.

[10] A.S.A.E.—Manager, motorcoach and truck division, Bendix Brake Co., South Bend, Ind.

[11] M.S.A.E.—Assistant chief engineer, chassis spring division, Detroit Steel Products Co., Detroit.

[12] M.S.A.E.—Chief engineer, Multibestos Co., Detroit.

[13] M.S.A.E.—Sales promotion engineer, International Motor Co., New York City.

H. D. CHURCH:—It is difficult to figure the cost comparison per mile of operation between the unlined metal-to-metal and the more conventional fabric-lined shoe on a direct basis. There is no question that in many operations the increase in tire cost due to the heating from the fabric-lined brake is entirely too high, although this is not true in most motorcoach installations.

For the last 1½ years we have been shipping motorcoach chassis equipped with metal-to-metal rear-wheel brakes and 20-in.-base tires, and have had no tire trouble due to heat generated by the brakes. These brakes were of the metal-lined shoe and drum type. In average city service the shoe liners have to be renewed in 15,000 to 20,000 miles, and the drum liners in 50,000 to 75,000 miles. These figures give an idea of what can be expected in service from the metal-to-metal construction.

In any motorcoach design passenger safety is the prime consideration and cost must be secondary. It is our opinion, based on many years of motorcoach manufacture, that for high-speed, heavy vehicles, the metal-to-metal brake with some form of servo actuation is very much safer than any other construction available today. This is true primarily because there is the least variation in braking effect due to water or lubricant on the brake-drums. We believe that for such vehicles the metal-to-metal brake is in most cases less expensive per mile than the fabric-lined brake, consideration being given to tire-cost increases which are caused in many operations by the difficulty of dissipating heat with the fabric-lined construction.

# Internal-Gear Four-Speed Transmission

## Discussion of S. O. White's Semi-Annual Meeting Paper[1]

ABSTRACT

INTENSIVE study of vibrations has had to be made by transmission manufacturers who were seeking a means of obviating the car noise that has become known as "high-speed rattle" and which makes itself heard through the transmission case, although it is caused mainly by periodic vibrations in the engine and propeller-shaft. Out of this study has come a revival of the idea of using a "fast" rear axle, that is, one having a low ratio between the engine and driving axle, whereby the rotative speeds of the crankshaft and propeller-shaft can be reduced greatly and the vibrations at a given road-speed consequently diminished.

To make the fast axle, or low rear-axle ratio, usable satisfactorily with present cars the four-speed transmission has been developed to provide more reduction in the transmission without making the steps in ratio between the several speeds too large. Four-speed transmissions have not been satisfactory in the past because of the noise caused by spur-gears and the wear of the teeth. Recent developments in internal gearing have shown that it is entirely feasible to design a transmission having internal gearing for third-speed that is both reasonably quiet and highly efficient. Such a design is illustrated and described in detail by the author, who explains its operation.

It is important to arrive at a rear-axle ratio that is the best compromise for all operating conditions. This should be as "fast" as possible yet avoid the necessity of much more gear-shifting than the public is accustomed to at present. A fairly close approximation of the best ratio to use, as made by plotting engine revolutions per minute against car miles per hour for a given size of tire and with a variety of ratios, shows that most cars produced today do not operate at the most advantageous torque of the engine horsepower-curve and that engines are run at too high speed. This causes excessive periodic vibrations, noise, wear of the engine and unnecessary fuel and oil consumption. The relation of various axle ratios to engine horsepower and torque, in third and fourth-speeds, for two cars of different size, are shown in charts and discussed at some length.

In the discussion, members favoring retention of three-speed transmissions advocate securing the desired results by more powerful engines and less gear reduction than in present cars, calling attention to some new European cars with three-speed transmissions. Others who are favorable to the newer four-speed transmissions develop points about weight distribution, gear-shift arrangement and bearing mountings, and outline an ideal four-speed transmission.

C. S. CRAWFORD[2]:—The first thing that strikes me as being essential about a four-speed transmission is to have the sales department educate the operators of cars to shift gears. If we had to do a little more of that in our three-speed cars we would not be thinking so much about the four-speed now. If we spent on the betterment of the three-speed transmission some of the money that is being expended on four-speed transmissions we should get a much better three-speed transmission and it would certainly be a much simpler design than the four-speed.

### WEIGHT DISTRIBUTION

ROBERT LAPSLEY[3]:—Our experience with high-speed rattle has paralleled Mr. White's as to both its causes and the steps taken to eliminate it. We firmly believe that a transmission somewhat along the lines that Mr. White has described will be the main remedy for this trouble.

Since the tendency in car design is undoubtedly toward lighter cars, we must add little weight; and, as the engine supports are already heavily loaded, we must concentrate the added weight as close to these supports as

[1] The paper with illustrations was printed in the June, 1927, issue of THE JOURNAL. The author is chief engineer, Warner Gear Co., Muncie, Ind.

[2] M.S.A.E.—Chief engineer, Stutz Motor Car Co. of America, Inc., Indianapolis.

[3] M.S.A.E.—Chief engineer, Detroit Gear & Machine Co., Detroit.

possible. We should have a transmission slightly heavier than the present three-speed transmission but with little additional overhang, somewhat more chunky than a conventional spur-gear four-speed unit but of no greater length. This ideal transmission should be of such a design that on high speed or direct drive no gears will turn. In the second high, as the term has been established for designating the internal-geared speed, the most desirable construction is such that only the internal-gear-drive parts will rotate, all other gears in the transmission being disengaged. To avoid making the other speeds in the transmission of such an expensive design as the internal-geared-drive second speed, low speed and reverse would be as near as possible to usual three-speed practice.

Mr. White has pointed out the fact that there will undoubtedly be a slight increase in the gear-shifting, and that, to make this possible without antagonizing the public, the operation of the shift between the direct and the internal-geared speed must be made extremely simple and positive. Parts that must change their rate of rotation at the moment of shift should be made as light as is practical. One group of men believe that this shift should be accomplished by positive-engaging clutches, while there will undoubtedly be a considerable number who believe with Mr. White that it should be by a smooth and easily made engagement.

### A Proposed Gearshift Arrangement

The arrangement of the shift is also open to discussion. As there are several millions of people in the United States who have become used to the three-speed transmission and its simple shift-arrangement, it might be best to keep the shift as nearly like the present standard three-speed-and-reverse shift as possible. Otherwise the four-speed transmission will be penalized by the fact that it is necessary to provide a latch of some type to prevent the engagement of reverse when the operator intends to shift into one of the forward speeds. It is possible to arrange the shift so that the standard three-speed shift will be retained and a latch provided for the speed that may be designated as "emergency low." The position of this speed in the shift should be such as to allow, when shifting from the customary low-speed, a motion such as we now have when shifting from low to second in the three-speed transmission. This type of shift should eliminate much of the sales resistance to cars equipped with four-speed transmissions. Any driver who is accustomed to a standard three-speed transmission would be able to get into a car with such a shift and readily operate it both forward and backward even though he did not know that it had four forward speeds. The only operation he would not be able to accomplish is that of starting in the extreme-low speed.

There will be a number of opponents to this arrangement of shift, but after they have operated a car equipped with such a shift they will appreciate its many advantages. It retains all the ease of shifting in straight-through motions and provides a shift with which a car may be left in any garage or parking lot where three-speed transmissions are well known, requiring no instruction of the attendants in the operation of the car.

What happened to a prominent man who drove an experimental car equipped with a four-speed transmission illustrates the value of this form of shift. He parked the car in a garage in a small town in northern Michigan, neglecting to inform the garage attendant as to the operation of the reverse. Upon returning to the garage he found that the attendant, while trying to locate the reverse shift, had bent and broken off the lever.

With a standard three-speed shift, drivers of cars are able to rock the car backward and forward in an emergency by quickly shifting from reverse to low and vice versa until they get the car out of a hole or pull through snow and ice. With the proposed shift having an emergency low this can be done, only the shift will be from reverse to second speed instead of to low speed. With the reverse controlled by a latch only experienced drivers are able to accomplish the shift in time to make the car rock back and forth.

It has long been recognized as desirable to build any unit in such a way that its component parts will have the strength and load capacity balanced so that all parts will have equal life. We all believe that the greatest amount of driving will be done in top speed; therefore the splines or clutches that transmit the torque of this drive should be made as large as possible to prevent excessive wear at this point.

In consideration of these facts our ideal four-speed transmission would possess the following features:

(1) Direct drive, a straight shaft with no gears in mesh
(2) Internal drive; no other gears in mesh
(3) Second-speed, called low, spur gears
(4) Reverse, spur gears
(5) First-speed, called emergency low, spur gears
(6) Standard three-speed shift with an emergency-low shift added
(7) Only slight increase in weight and length over three-speed transmission
(8) Large splines or clutches for transmitting the torque of the engine on direct drive
(9) Transmission case must be rigid

I have purposely left until the last any mention of ratio of either the transmission or the rear axle. Ratios in the transmission must be great enough to give a desirable reduction of engine-speed. The ratio in the rear axle will have to be worked out for each individual car. It must not be too fast an axle as this will tend to destroy the acceleration of the car, and it must not be too slow as thus the benefit of slowing down the engine will be lost. Mr. White, in his discussion, has covered this very clearly and I have mentioned it only because the success or failure of the whole idea depends on the correct selection of ratio for the rear axle.

W. G. Wall[4]:—The four-speed transmission is not acceptable unless the third speed can be made quiet. Mr. White has succeeded well in doing this. I do not mean to say that third speed is as quiet as direct drive, but at least the internal gear seems to give a fairly quiet third-speed; but it takes up much room and is very expensive to build. The question, I think, is not so much whether we want three-speed or four-speed transmissions as whether we can make the four-speed transmission sufficiently quiet on third speed to be acceptable.

Chairman E. S. Marks[5]:—We are all agreed that it is desirable to make cars run 50 m.p.h. with the same degree of comfort and smoothness that they formerly had at 35 m.p.h. The four-speed transmission is one of the methods by which we hope to accomplish this. There may be other ways.

[4] M.S.A.E.—Consulting engineer, Indianapolis.

[5] M.S.A.E.—Chief engineer, H. H. Franklin Mfg. Co., Syracuse, N. Y.

### Three Speeds on Large European Cars

F. E. Moskovics[6]:—What was generally admitted last year to be the best car in Europe had a three-speed transmission. On the high-priced Hispano there was an extremely flexible engine and a three-speed transmission. On the smaller Hispano, which is not so successful, there was a four-speed transmission. The same is true of the Bugatti and of several English cars. Transmission requirements are determined by engine performance. The question is, first, what high-speed activity is wanted. That can be determined by acceleration tests on high speed, say from 5 to 50 m.p.h. If there is the desired high-speed activity, which means also hill-climbing activity, what will the characteristics of the engine permit on high speed? I agree that a smooth 50 m.p.h. is needed. A car that will run 80 m.p.h. and has good high-speed acceleration will have a smooth 50, and if this can be obtained with a one-speed or a two-speed transmission that is better than using more speeds.

William Ernest England[7]:—Mr. White made the statement that after taking all the gears out of the transmission it still had the rattle in it. My experience is the opposite. We removed the secondary shaft from a car that had a bad high-speed rattle at 35 or 38 m.p.h. and with the gears entirely out of mesh the noise was absolutely gone. With the secondary shaft in place we were able to do away with the rattle by putting a mechanical dampener into the clutch.

Chairman Marks:—Mr. White said that the mechanical dampener in the clutch is one of the things with which we should become familiar. I had an experience with a very bad medium-speed rattle. We introduced a flexible member into the driven-disk assembly of the clutch and it did away with that trouble. It not only helped in that particular speed range, but it helped materially through the whole driving range. The idea has since been adopted on cars of several other makes.

### Preliminary Experiments

F. C. Thompson[8]:—The Morse Chain Co. started development of four-speed transmissions about 10 years ago. Its silent-chain transmission was satisfactory as to quietness and durability but not as to weight and cost. Realizing the advantages of the four-speed transmission, the company's engineers continued their research and development work to produce a transmission having the two highest speeds sufficiently quiet so that either would be entirely satisfactory. Work has been under way for more than 3 years on the Thompson four-speed internal and external-gear unit, which can be used as an over-speed with direct drive on third speed, or an under-speed with direct drive on fourth speed. This unit can be used in connection with a standard transmission or incorporated in a special design.

Direct drive on third, as shown in Fig. 1, has the advantages of reduced propeller-shaft speed and lower axle-ratio, but, if the car is powered so it can be handled satisfactorily in traffic with a total reduction-ratio of approximtely 3½ to 1, direct drive should be on fourth speed and the internal and external gearing should be used for the third speed, to give quiet acceleration and hill climbing. Many modern cars have rear-axle ratios of about 5 to 1 to give sufficient acceleration and hill-climbing ability. With such a rear-axle ratio direct drive on third speed is more desirable. Direct drive should then be used for all ordinary driving up to 30 or 35 m.p.h., with the overspeed for country driving. With a step-up of 7 to 10 the fourth speed reduction of such a car is 3½ to 1, reducing the engine-speed 30 per cent for the same car-speed and considerably increasing the maximum speed of many cars. This arrangement retains the present high-speed engine and cuts down its operating speed. At speeds of 30 m.p.h. and more, fuel consumption and wear of engine parts are reduced and the operation of the engine is smoother, owing to the reduced engine-speed. Also, vibration at high speeds is reduced because synchronism is prevented when the crankshaft and the propeller-shaft rotate at different speeds.

Having equipped a great number of cars of different make and size with four-speed transmissions, we have reached the conclusion that it is generally preferable to use an overspeed transmission unless the engine size is to be increased. One thing in favor of the over-speed is that the internal and external gears operate more quietly at the higher speeds. Whether the indirect high-speed is third speed or fourth, it must be sufficiently quiet so that it will be used at the speeds for which it is intended. Quietness is largely dependent upon rigidity, which means large shafts that will not deflect appreciably and large bearings rigidly mounted. The size of the internal and external gears also is of importance, the gears that operate most quietly being so large that they look entirely out of proportion to the rest of the transmission.

### Will Americans Shift Gears?

C. D. McKim[9]:—With European cars changing to three-speed transmissions it seems more than ever a hopeless task to educate Americans to shift gears so often and with the same ease as do Europeans. With improving roads and increasing long-distance traveling, and with weight and economy unimportant, the three major essentials in car performance may be arranged in this order of importance:

(1) Comfort
(2) Acceleration, including hill-climbing and general performance
(3) Economy

The sensation you experience when coasting down a hill with the clutch disengaged and the engine idling is due largely to the low engine-speed. Transmissions, differentials and axles are not sources of power. There is only one sensible and mechanical way to get that comfortable coasting sensation and that is to put more creative power under the hood. A little additional power can be had with little added expense, perhaps with less expense than a special transmission.

Thomas L. Fawick[10]:—Twenty years ago engines were so large that they were clumsy and expensive to operate, but now they are smaller. Putting in a big engine does not cut down engine-speed, and that must be reduced to get comfortable driving-qualities on the country roads. On the smooth roads, where it takes very little horsepower to pull a car, it is a plain waste of energy and of material to put in a big engine and run it at high speed, even though the engine runs smoothly and quietly. If you put larger engines into present cars, other parts must also be larger and the result is larger cars instead of reduced engine-speed. Cut down the speed of the engine and with proper driving-ratios it will still pull the car and negotiate any ordinary hill.

[6] M.S.A.E.—President and general manager, Stutz Motor Car Co. of America, Inc., Indianapolis.
[7] M.S.A.E.—Chief engineer, F. B. Stearns Co., Cleveland.
[8] M.S.A.E.—Manager Detroit branch, Morse Chain Co., Detroit.
[9] Sales and advertising manager, Hercules Motors Corporation, Canton, Ohio.
[10] M.S.A.E.—Racine, Wis.

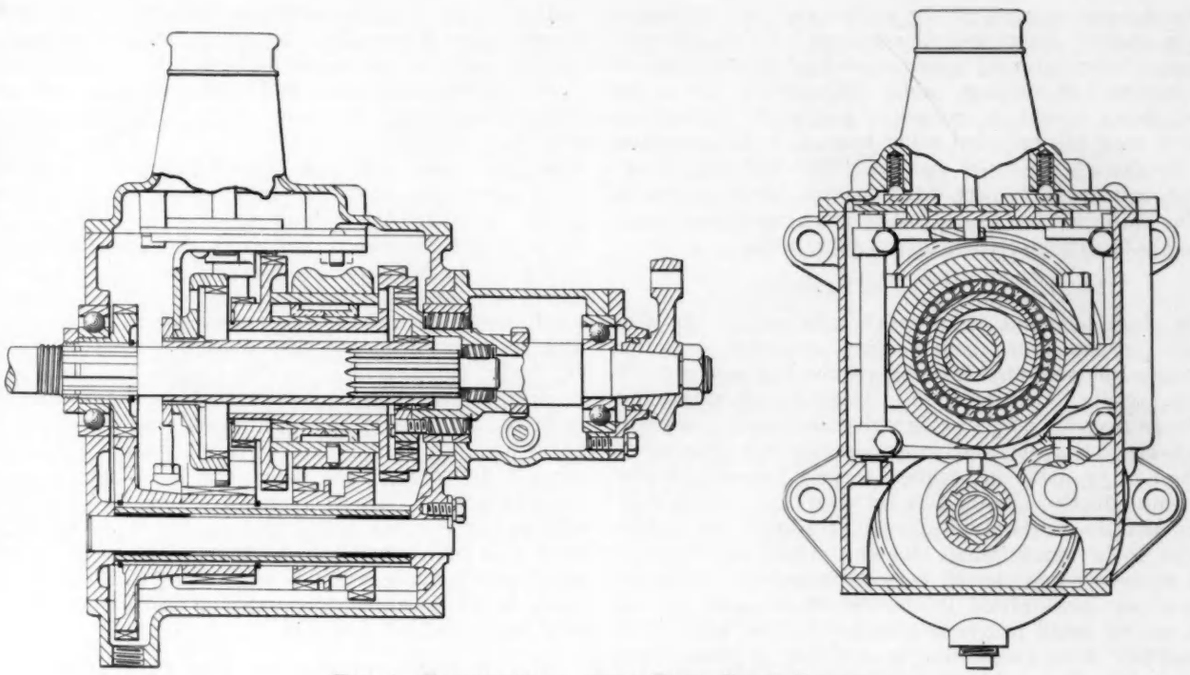

FIG. 1—TRANSMISSION WITH THIRD-SPEED DIRECT
Compact Morse-Design Transmission with Internal-Geared Over-Drive for Fourth Speed

There are not many men here who have driven cars over hills in all kinds of conditions with a car that had two quiet driving-ranges. If you ever drive such a car you will no sooner drive a car with a three-speed transmission than one without a self-starter.

MR. CRAWFORD:—We find that one 300-cu-in. engine develops a maximum output of 75 hp. while another of the same size develops 115 hp., over 50 per cent more than the first. Considering the wonderful output of racing engines, it seems possible to get materially better engine-performance than we have, and at the same time to make engines smaller and lighter. I appreciate the advantages of the four-speed transmission; perhaps that is the solution with an under-powered engine; but I believe we should work more for simplicity and for greater output from our engines.

### MOST DRIVING IN FOURTH SPEED

G. C. MATHER[11]:—Until a long trip has been taken in a car equipped with a quiet, efficient four-speed transmission it is impossible to conceive the comfort and enjoyment of such a ride, due largely to the low engine-speed with a comparatively high vehicle-speed.

Many believe that the fourth speed can be used only in the open country or at speeds from 25 m.p.h. up, which is not correct. We find, and recommend to our customers, that at any speed above 10 m.p.h. fourth speed can be used unless extremely rapid acceleration is desired.

My company has had in production the model equipped with the four-speed transmission since the end of February and, according to reports we receive from our dealers throughout the Country, this device is being received by the public with even more enthusiasm than we had anticipated, which, in the last analysis, is the most important consideration.

One opinion mentioned in the discussion was that the four-speed transmission is unnecessary. Most car manufacturers establish the minimum performance, selecting some more or less well-known grade, such as Uniontown Hill, which the car must be capable of climbing in direct drive at a certain minimum speed, or with a fixed minimum elapsed time. In a vehicle capable of climbing such heavy grades on direct drive, the engine is running entirely too fast for 90 per cent of the driving.

Others opposed to the four-speed transmission advocate a larger engine and a higher-geared axle to attain the same smooth running. No matter how large the engine and how satisfactory the performance, it can be greatly improved by the addition of the four-speed transmission.

If we did not have the four-speed transmission in our own car this model would have a rear axle geared 4.8 to 1.0. With the four-speed transmission it is geared 3.69 to 1.00 in direct drive or fourth speed, which is used more than 90 per cent of the time even in hilly cities such as Pittsburgh. We also have the advantage of a quiet-running third-speed, geared 5.25 to 1.00, that can be used for the occasional conditions when extremely rapid acceleration is desirable, or when extraordinarily heavy grades are encountered.

It is my opinion that within the next 2 years the use of the four-speed transmission will spread throughout the industry, and that a car without it will be almost as rare as one without electric lighting-equipment.

C. E. F. AHLM[12]:—Mr. White has given us an instructive contribution to a subject which is now uppermost in the minds of the automotive engineers and, I dare say, of the public as well. The desirability of a fourth speed has long been recognized but no real progress was made until the admirable qualities of the internal gear were brought out by study and practice.

A fourth speed, whether built into the main transmission or an auxiliary, whether an over-gear or an under-gear, to be commercially acceptable must be quiet, effi-

[11] M.S.A.E.—Chief engineer, Paige-Detroit Motor Car Co., Detroit.
[12] M.S.A.E.—Manager and treasurer, Continuous Torque Transmission Co., Cleveland.

cient and easy to manipulate under any conditions of road or speed. Quietness is attained by the inherent features of the internal gear, important considerations being manner of cutting, tooth difference between the mating gears, backlash, pressure angle and mounting. We have used full ball and roller-bearing mountings and plain bearings with equal success. The ball and roller-bearing transmissions are heavier and more expensive than the plain-bearing units, but in most cases they make it possible to dispense with forced lubrication.

### Advantages of Plain Bearings

With plain bearings it is possible to reduce the size of the internal gears considerably, thereby reducing their pitch-line velocities as well as the size and weight of the entire transmission. Though plain-bearing mountings may be somewhat greater in length, they are simpler and cheaper, and more certain to give quiet operation if properly lubricated, largely because of the cushioning effect of the oil film between the rubbing surfaces. Medium engine-oil forced through the plain bearings by a small pump drawing clean oil from a baffled reservoir has proved very satisfactory. In some instances we have found it convenient to pass the oil over a set of small magnets attached to the bottom of the reservoir, thus preventing steel flakes or scale from the gears from being forced into the bearings.

Efficiency of the fourth gear is of utmost importance, and here again the internal-gear assembly is superior to spur gears. If correctly made and mounted there should be less than 2½ per cent loss in a set of fourth-speed internal-gears.

The shift between third and fourth gear, or vice versa, in a four-speed transmission must be easy, quiet and positive. The operator should and will use the extra speed frequently to secure the fullest benefit of the combination. It is true that the difference in ratio between third and fourth speed is less than between second and third in the conventional arrangement, but still we find that the ordinary method of gear-shifting is impracticable, and after much study I have devised a clutching arrangement[13] in which the permanently mounted rotative members are clutched together by a hollow sliding clutch provided with a set of internally-projecting teeth at each end. One set of these teeth is in constant mesh with one of the rotating members while the other set is made to engage with teeth hobbed on the member to or from which motion is to be transmitted. From this second set of teeth in the sliding clutch-member is cut out every other tooth, or two of every three teeth, depending upon the characteristics of the shift under consideration. The mating member has an equivalent number of external teeth cut away for a distance of 7/64 to ⅛ in. This introduces into the shift a definite time-element, resulting in an easy, positive and quiet shift.

### Short Demonstration Not Conclusive

S. O. White:—Most of those who object to the four-speed transmission have had little or no actual experience with it. To appreciate it fully one must spend a day, or perhaps a week, in both level and hilly country and in city traffic, with both good and bad roads.

Since the advantages come largely from the fast axle, the obvious question is, Why not provide the fast axle with a conventional three-speed transmission? The answer is another question: Why is this not being done generally at present, since there is nothing new about a fast rear axle?

[13] See The Journal, February, 1927, p. 250.

It is quite possible to use a fast axle if a sufficiently large engine is provided. Some of the speakers have said that there is no particular objection to a large and powerful engine, and have cited examples of certain prominent foreign cars that are adopting three-speed transmissions and leaving the four-speed. These are in the heavy-car class, and there is a certain limited market for very large and powerful cars among a small class of people to whom the amount of gasoline they buy is of no particular moment; but, with the increasing density of traffic, the small car is coming constantly to the front. There are several notable examples of new, high-grade small cars. With more general extension of paved roads and the improvements that are constantly being made in body construction, spring-suspension and chassis mounting, the riding-qualities that were formerly found only in the heavy, long-wheelbase cars are gradually being acquired by the smaller cars. Except for a limited market, the day of the large and heavy car is passing. Furthermore, it is well known that the trade-in value of a large car is much less in proportion to its initial price than that of a small car, because people buying second-hand cars usually are not willing to pay for the large tires, heavy gasoline consumption and generally expensive upkeep of a heavy car.

### A Powerful Engine Not Equivalent

There are certain qualities in the four-speed internal-gear transmission which give results that cannot be obtained with a three-speed transmission. A fast axle with a very powerful engine only partially accomplishes the results desired. As mentioned in the paper, the second speed of three-speed transmissions is a compromise, too slow for good hill-climbing work. It is also so noisy that drivers object to using second speed even so much as it should be used with the present cars. It has been suggested that the transmission manufacturers should put more effort into refining the present transmissions. Transmission manufacturers and car builders with thousands of engineers and gear experts have been working on this problem ever since transmissions were first brought into use, with the result that the transmission of today, though far from perfect, is a considerably refined product. Engine builders have so much trouble to produce quiet timing-gears that many have adopted silent-chain drives.

While the internal-gear drive is not entirely silent, it is more nearly silent than any spur-gear drive. With it one can now drive up or coast down a long mountain road with about the same amount of noise as from one of many rear axles and, now that attention has been focused upon it, improved methods of mounting and of gear cutting will make the internal drive still more nearly silent.

It is quite feasible to produce a three-speed transmission with an internal-geared second-speed, but as a large part of the expense connected with this design is chargeable to the internal-gear mechanism itself, the difference in cost between such a transmission and the four-speed is not very great, while there are many advantages in favor of four speeds.

Shifting between third and fourth speeds has already been mentioned. If shifting between second and third could be made easier in a three-speed transmission it would be worthwhile, provided it did not entail too much added mechanism. In the design under consideration the mechanism for shifting between third and fourth is very simple and the shift is so easy that the driver almost unconsciously makes more use of the next-to-top

speed than he ever did before. It would be worthwhile if we could get him to do the same with three-speed transmissions.

GEARSHIFT POSITIONS

A suggestion has been made to arrange the four-speed control so that first speed would be considered an emergency low and be latched out, having three forward speeds and a reverse in the same arrangement as in a standard three-speed shift. At first thought this is an attractive idea and quite possible to accomplish, but further consideration makes it seem questionable whether such an arrangement would be so desirable as claimed.

Ever since four-speed transmissions have been built for either passenger-cars or trucks it has been customary to latch out the reverse. Garage and parking-lot attendants are familiar with the usual truck-shift, and when a four-speed car is left in their care they expect to find reverse protected by the latch on the control lever. In the course of about 2 years' experience with many different cars and garages all over the Country we have found no particular difficulty from this. We are afraid there would be difficulty if the first speed were the one latched out, because the garage attendant would expect that to be reverse and would perhaps find out too late that it was a forward speed. First speed is used fully as much as reverse, and second gear is too high to operate satisfactorily in trying to rock the car out of deep snow, mud or sand, so there would have to be a cross-over between first and reverse for this.

The present generally used three-speed arrangement is not ideal because in high the control lever is in the way of the knee of the occupant of the right-hand seat. With the arrangement that we have, fourth speed finds the control lever in a forward position, close to the instrument board and entirely out of the way. The presence of a thumb latch on the four-speed transmission reminds the driver of the control arrangement.

It would be very unwise to overlook or condemn the four-speed transmission on purely theoretical considerations without a very thorough try-out under a great variety of driving conditions.

# DIESEL-ENGINE DEVELOPMENT

AMONG the more important future motor-transport developments the general application of the Diesel engine to road-vehicle propulsion now seems more than probable. In view of its long success as an economical and reliable power producer for marine and stationary work, it is, in fact, surprising that it has not already become a competitor to be reckoned with in a field where the gasoline engine has gained so great a lead. Admittedly, the practical difficulties of adapting the Diesel engine to road-service conditions have been great enough to discourage the most intrepid designers when one considers the importance attached to weight and size reduction. That great progress has been made in this direction, however, can be gathered by the fact that Diesel and semi-Diesel engines suitable for aircraft propulsion have been constructed.

Credit for the development of light high-speed engines operating on the Diesel principle lies chiefly with Continental and American designers and manufacturers, and there appears to be some danger of British engineers being left behind in this particular field. Already in Germany a number of Diesel-engined motor-trucks have been running under ordinary service conditions with decidedly encouraging results, the power units demonstrating not only degrees of flexibility and reliability comparable with those of the ordinary gasoline engine, but indicating the considerable economy possibilities of using fuels materially cheaper than the lowest grade of motor fuel. It is, of course, in this latter connection that the Diesel engine holds principal attraction for the motor-transport user, particularly in undeveloped countries overseas, where suitable fuel can be obtained from local sources and where motor fuel is procurable only with difficulty and at prohibitive cost.

For fast light-load vehicles the electrically ignited carbureter-engine is not likely to be displaced, at least for many years, but for large and medium-capacity machines we believe the advent of the Diesel engine has been brought very much nearer by results recently achieved on the Continent. In the direction indicated there is yet a very wide field for experiment and research, and it is to be hoped that realization of the prospects by British manufacturers will assure at least their equal participation in the work to be done and its reward.—*Motor Transport* (London).

# PANAMA CANAL OUTGROWN

REVIVED discussion of a trans-Isthmian canal at Nicaragua is a reminder of the stupendous industrial development and transportation demands of the Country in recent years. When the Panama Canal was opened, few people believed that, short of many decades, it would pay its way. Inside of 12 years, it is paying for operation, interest on the investment and a big annual profit to the Government. Now its business is in sight of exceeding the canal's capacity. Whether a new ditch shall be built in Nicaragua or the Panama Canal shall be rebuilt to increase its capacity, it has suddenly become seriously doubtful whether either plan can be executed soon enough to avoid congestion.

By far the largest factor in the canal's business is one which nobody anticipated at all when the canal was started, namely, the transport of petroleum. While this particular business has grown so fast that it threatens to exceed the canal's capacity, it has not only taken no tons away from the railroads, but has actually added enormously to their freight business.

Rough estimates place the cost of the canal at Nicaragua around $1,000,000,000; Panama cost about $400,000,000. For approximtely $125,000,000 new locks and dams could be constructed which would greatly increase the capacity at Panama. But the argument against putting all the eggs in one basket is potent, because both industrial requirements and security of naval movements are widely believed to dictate the policy of having two canals rather than a single greater one.—American Petroleum Institute News Bulletin.

# The High-Performance Small Car

## Discussion of T. J. Lítle, Jr.'s, Pennsylvania Section Paper

STATING his belief that we are on the threshold of the most revolutionary development that the industry has ever witnessed and remarking the stages through which the industry has passed, the author mentions the apparent settling down of the industry to the refinement of its product and says that comfort and performance now dominate the American mind in its demand for better automotive transportation. He believes that America will not copy the foreign car, although the good features of the many foreign designs are recognized; but he predicts that the influence of racing-engine development will be felt and that the present ideas of engine building will be entirely changed on account of the wonderful performance of racing cars, the result being a small high-speed high-power passenger-car engine operating at a speed of at least 4000 r.p.m. The author sees the completed evolution eventually of a small car that will be equipped just as luxuriously as the finest car on the American market, one that will be fitted with a special custom body but will be in no sense cheap.

Regarding the present types of car, the author prophesies, first, the elimination of the very cheap hard-riding car of obsolete design; second, the discarding of the medium-price car in about the same class; and, third, that the new engineering school will create the fine small car. Although he recognizes that the American and the European problems are very different, he says that Europeans are coming gradually to the American idea and that the European public eventually will buy millions of new cars of a type that will provide comfort, luxury and maximum performance. He is convinced that the industry can learn much from the designers of racing cars and that supercharging and the use of high-compression fuels will be included in future improvements. He sees an era of production of many fine small and medium-size cars and the formation of numerous companies for that purpose.

European practice in small cars was treated in the discussion, including their performance on American roads and the attitude of American drivers toward such cars.

H. L. BROWNBACK[2]:—I have been particularly fortunate in living part of the time in Paris as well as part of the time in the United States. As I know a great many foreign automobile designers rather intimately and speak their language, being of their race, they probably talk to me a little more easily than to the average person who goes over there. In addition to this, about a year ago I bought one of the very latest types of French car, not what they call a small car but a medium-sized car, brought it to the United States with me last spring, and have been driving it here this summer in conjunction with an American car, to find out how the two cars compare.

One of the principal features of the French car is the small engine. As Mr. Lítle has said, they had to get performance from the small engine so they began by increasing the speed. Most of the L-head engines run at 3600 r.p.m., some at 4000 r.p.m. and others at 5000 r.p.m., so the question of lubrication is brought to the fore. The vehicles in which these high-speed engines are installed must push very easily. I think that is where the American chassis is deficient. The average American chassis with roller bearings and not-too-careful alignment, using heavy grease in a great many parts, does not push easily. If I jack up one rear wheel of this little French 10-hp. car, using castor-oil in the rear axle, the wheel will turn until the tire valve is at the bottom.

### DETAILS OF EUROPEAN CARS

The use of roller bearings is almost unknown in the small, easily pushed foreign car. Ball bearings are used in the front and rear wheels and on either side of the differential. The springs of the chassis hug the road rather closely, and enclosed propeller-shafts and shock-absorbers are almost universal. Every car uses shock-absorbers.

Another thing that is practically universal, even with the small 5-hp. cars, is the use of four-wheel brakes. They are absolutely necessary over there. The four-wheel brake that they use is radically different from the American. In practically every case it is an internal-expanding mechanical-brake faced with very hard brake-lining, the brakes dropping away from the brake-drums to allow an absolutely free-turning brake-drum when released; and, in the majority of cases, the brake-drums are steel forgings. The trend is toward the light six-cylinder engine, not so much for home use as for export sale.

The European chassis is more flexible than the American chassis. Foreigners have gone in more for the sporting type of car. When closed bodies are put on, they have a tendency to rattle badly on the cobblestone roads in France, even on the fine cars.

Another thing that is causing some trouble is front-wheel shimmying with balloon tires. The shimmying has not been so bad there as in American cars because the engines are so small that if the balloon tires are run under-inflated, as they are in this Country, all the power is consumed. To a great extent that has prevented shimmying.

### A FRENCH CAR ON AN AMERICAN HILL

On the road between Philadelphia and Reading, above Norristown, is a hill called Providence Hill which has approximately a 15-per cent grade about 1/3 mile long. If I reach that grade with my little 10-hp. car when running at about 35 m.p.h., with 35-lb. per sq. in. air-

[1] The paper, which was presented at a meeting held at Philadelphia on Nov. 9, 1926, was printed in the December, 1926, issue of THE JOURNAL. The author is an engineering production consultant, Detroit; also vice-president in charge of engineering, Copeland Products, Inc., Detroit.

[2] M.S.A.E.—Managing partner, Henry L. Brownback & Associates; vice-president and general manager, Brownback Motor Laboratories, Inc., New York City.

pressure in the tires, which are about 5¼ in., I can go up the grade carrying two passengers with very little loss of speed. At a speed of 25 m.p.h. I can just about go over the hill. At 20 m.p.h. I go over nicely in second speed. If I strike the grade at 15 m.p.h. I must go into low gear; then, when the engine-speed reaches 3000 r.p.m., I can change to second gear and keep the speed up to 2500 r.p.m., but it is necessary to go through that procedure. If there is only 15 lb. of air in the tires I have some difficulty in getting over the hill. The gear-ratio is about 4½ to 1.

The steering-gears are high-grade, irreversible and quick. The steering-wheels are more or less flexible. A spring-steel-spoke wheel is made with a very flexible thin rim. Ordinarily, steering-wheels are made of celluloid with an aluminum rim having little elasticity.

With its extremely high-speed engine a little French car is very fast, although it looks like a cut-down "flivver." L-head engines have a 5½-to-1 compression-ratio and develop 33 hp. at 4000 r.p.m., and the gear-ratio is 3½ to 1.

Lubrication is one of the things with which I experimented after coming to the United States. With ordinary mineral-oil of medium viscosity and high quality I can get about 96 km. per hr. (60 m.p.h.). Using castor-oil I can get about 115 km. per hr. (71 m.p.h.). To maintain high speeds, castor-oil must be used. The engine will not operate with ordinary gasoline. It is dead; it simply detonates. The car weighs 2500 lb.

Another thing in which the small French car differs from American cars is the tread. It is about 53 in. on a 112-in. wheelbase, as I recall. They have found, by careful experimentation, that there is a certain relation between the tread and the wheelbase at which the vehicle is most stable. It was found on some cars with very short wheelbases that increased stability is obtained if the rear tread is made slightly wider than the front tread. This is noticeable and a very important point with small high-speed cars. After more concrete roads have been built in this Country I think the 56-in. tread will not be so necessary as it has been.

About the only other thing that is very noticeable in foreign cars is the use of a stiff clutch. The clutch has a single plate clamped between two surfaces, with asbestos composition on both sides, generally engaging with six or eight springs, and its operation is exceedingly stiff as compared with American designs.

The main thing, as I have tried to outline briefly, is the extremely easy running of the chassis. The high-speed high-compression engine necessitates a little better manufacturing than we have been getting in vehicles in this Country. In the European factory, girls who receive about 4 francs per hr. are used on production work. A skilled mechanic, who wears wooden shoes, gets 5 or 6 francs per hr.

B. B. BACHMAN[3]:—Another thing might enter into this discussion: Whether the American public as a whole will accept a development such as this that has been very acceptable abroad. The small car certainly seems popular in France where it is seen in large numbers.

C. O. GUERNSEY[4]:—Every year we see in the trade papers pictures and specifications of composite cars of various classes. Mr. Lítle has spoken of three classes of light-weight car: (*a*) the light-weight, high-efficiency, luxurious car that must sell at a high price, comparing with the present cars of the $2,500 class or higher; (*b*) the car having a high-efficiency, high-speed engine, moderately well equipped, which would compare with cars in the present $1,500 class; and (*c*) the cheaper high-efficiency, six-cylinder car, which is intended to replace the present four-cylinder car selling at from $600 to $1,000.

I should like to ask Mr. Lítle to draw specifications for those cars; telling not how large the engines should be but what the cars will weigh, their top speed, their acceleration, what kind of brakes they will carry, and giving his idea of the various types, with pictures of their performance rather than of the cars themselves.

### ONLY FOUR CYLINDERS NEEDED

H. C. GIBSON[5]:—I do not think we are here to criticize Mr. Lítle, although he seems to ask for it. We must first overcome the inherent generosity, or is it the spendthrift tendency, of the public of the United States, the motoring public at any rate, which is certainly willing to pay for comfort. Supreme comfort apparently is not secured in the cars that Mr. Lítle has described, although I believe we can get all the comfort we need in a light cheap four-cylinder car.

I believe that the six-cylinder car would not be so popular as it is today if the four-cylinder engine had less vibration. I do not think anyone minds the vibration produced by the push in the cylinder so much as vibration from other sources. As a matter of fact, a great many persons like to feel the pull of the engine. If the four-cylinder engine can be freed from objectionable vibration, that would help toward getting a satisfactory light car. The four-cylinder engine is lighter for its horsepower than is the six-cylinder engine, and that is why so many of them are used in the lighter cars.

We can obtain high engine-speed here. We have not developed automobile-engine bearings to the extent that we might to take care of the high speed, but it is accomplished in the motorcycle engine. These engines have beautifully ground hard-steel-bushed bearings that will last practically forever. Following the practice that seems now to be confined to the motorcycle would afford a fine opportunity of getting a very much lighter car. With a lighter car smaller tires can be used. Balloon tires are very cumbersome and involve the use of heavy rims.

Much more attention should be paid to the reduction of friction and the weight in the car, and longer springs should be used. American cars are difficult to push because of the excessive weight provided for comfort. Nevertheless I do not believe that America is going backward. The well-made light-car is already here.

### AMPLE PASSENGER-SPACE REQUIRED

MR. BACHMAN:—One phase which it seems to me has not been brought out in the specifications that Mr. Guernsey asked Mr. Lítle to give is, how many passengers will the car carry? It seems to me that this is a very pertinent question. I do not believe that the human figure will change very much or very rapidly. Consequently, the space that must be provided for passengers will have a powerful influence on what the ultimate size of the vehicle will be. For instance, take the matter of tread, which has been mentioned. If two persons are to be carried on the rear seat of a low-hung body between balloon tires it seems to me that the matter of tread is nearly settled. The same is true of the fore-and-aft dimensions. Many of us have long legs and we have a habit of resting our weight on the back of the neck when

[3] M.S.A.E.—Engineer, Autocar Co., Ardmore, Pa.

[4] M.S.A.E.—Chief engineer, automotive car division, J. G. Brill Co., Philadelphia.

[5] M.S.A.E.—Consulting engineer, Engineering Developments, Inc., Philadelphia.

we sit. This has an influence on the fore-and-aft dimensions. Perhaps we can reduce the engine to such a size that it will not take up much room, but the American public seems to want something in front of the passenger space. So it seems that the dimensions of the car are rather definitely controlled by considerations of comfort and appearance.

MR. GUERNSEY:—In addition to asking for a specification, I should like to have Mr. Little sell me that car. Will it be sold on the basis of comfort, performance, low cost of operation, long life, or convenience in parking? What is the feature that will make it possible to sell the car to the public in competition with cars such as we have? If it is on the cost of operation, which is one of the obvious things that must be considered in small cars, a $2,500 car costs about $0.15 per mile to run; and of that amount we should allow about $0.08 for depreciation. The highest charges: garage, cleaning, interest, insurance, license, gasoline tax, depreciation, and interest on the investment, will amount to about $0.11 of the $0.15. If the car is to sell for the same price, there will be little saving unless the depreciation or some of the fixed charges can be reduced. The savings that can be made on fuel, tires and the like amount to only $0.01 or $0.02 per mile.

### Advantages of Electric Transmission

W. J. MAYER[6]:—The trend of automobile design seems to be toward the tremendously high-speed engine, involving many difficulties in the accurate dimensions of the pistons and other parts because of the very fine machining that is necessary to secure proper balancing. It is a question whether some advantage cannot be gained by developing the electric transmission and using an engine-speed of possibly 2000 to 2500 r.p.m. A generator of this speed would not be excessively heavy nor bulky. By automatically cutting-out field resistance in the generator, or by shunting current around the fields, the engine-speed could be reduced at high car-speed. This would simplify the problem of lubricating bearings and gears.

CHAIRMAN E. W. TEMPLIN[7]:—Three men in my office recently were figuring what it cost to own an automobile because of the depreciation. One man owning a Dodge car figured the cost to be $0.90 per day; another man owning a Chrysler, and the third with a Cadillac, estimated the cost to be practically the same. If it costs the same to own any car, we may as well have what we wish; and possibly Mr. Little will produce it for us.

I noticed particularly Mr. Little's reference to light bodies. At the American Electric Railway Association convention in Cleveland in October, 1926, a street-car of large size was shown having its body made completely of aluminum. So far I have not heard of an automobile passenger-car body being made completely of aluminum.

MR. MAYER:—We have experimented with duralumin for body construction. In 1922 we built a phaeton body on a chassis of a prominent make. We wished to see how this metal could be handled in this sort of production work. The body was welded together, not riveted. This body has been in continuous use at our plant ever since without developing any serious defect. At present the manufacturing of duralumin automobile-bodies on a large production scale is out of the question. The all-steel body is heavier than one of aluminum but steel is preferable on account of the lower material cost and greater ease in handling.

[6] M.S.A.E.—Assistant chief engineer, Edward G. Budd Mfg. Co., Philadelphia.

[7] M.S.A.E.—Chief engineer, Six Wheel Co., Philadelphia.

### A Small Car Can Be Comfortable

T. J. LITTLE, JR.:—Regarding the comfort of these light bodies, I have said that the small car will be as easy riding as the finest car in America today. There is no reason that a small car cannot be just as comfortable as a larger car. Size does not necessarily regulate comfort as it does in the American car today. I have ridden in some small cars that are comfortable, and some that give the impression of sitting on a telephone book on the floor. The latter style of car would not be popular here, you may be sure.

It may be said that the three men who so carefully compared notes on the cost of operation of their three cars were engineers, and engineers sometimes make mistakes. But, to speak seriously, it is difficult to obtain an expression of that sort from ordinary men, because very few know what it costs to operate their cars. I doubt whether many of you here know exactly how many miles per gallon your own car will go.

I hesitate to specify exactly what these cars that I mentioned will weigh; my guess is that the small car will weigh about 1200 lb., will have 125 or 150-cu-in. piston displacement and seats for four passengers. I think the middle-size car will weigh about 2500 lb., will carry five passengers, and have an engine of 225-cu-in. displacement. The large car will not resemble these cars. I think the seven-passenger car will weigh about 4000 lb. and have an engine of about 400-cu-in. displacement. When the rich man buys the large vehicle, he will desire and demand in this the same performance that he can get in the small one; therefore he will demand a large engine. That is an entirely different demand to fill.

It has often occurred to me, and I suppose to others, that it is strange that foreign manufacturers have not come to this Country more to manufacture their product or to sell their cars. They have not sent many cars here, and some of us wonder why, if we are coming to the light car, they could not bring over some cars now and sell them. I believe some of them could do so.

### Performance Sells Cars

The paramount selling feature of the small car will be better performance. We want cars that will outperform those we are now driving, otherwise we will keep the old ones. Better performance seems to be the principal reason for buying a new car. Of course body style and beauty will always be important. I think we make much better looking cars in this Country than they do in Europe. In the automobile salons in New York City the American bodies are much handsomer than those of the foreign cars. We will buy better looking bodies and better performance every year.

So long as the cost of operation is not excessive a difference of a fraction of a cent per mile for operation will not be very important in this Country. We are not troubled by such high taxes as there are in Europe. Greater economy is desirable but that will not sell a car. Riding comfort will be very important, because cars that are hard riding cannot be sold in America.

The gasoline-electric combination, I think, will be too heavy for passenger-cars. It is smooth running, but too heavy and unnecessarily complicated.

### Exorcising Vibration

We all have used some barnyard engineering, otherwise known as common sense. In the development of the Lincoln car we had to do some of that kind of engineering. After the car was designed and placed on the

market it was found to have entirely too much vibration. At that time we had no chief engineer and I was in charge of the experimental department and research laboratories. We had a meeting at which all the engineers gathered in the president's office to decide what should be done to make the engine good. At least 11 or 12 engineers were drawn into that conference. It was decided that we would experiment with many different things that might cause vibration in the engine. I suggested that perhaps the 32-oz. cast-iron pistons or the connecting-rods were too heavy, or that the vibration was in the crankshaft, the camshaft, the vertical shaft, the fan, or in anything that moves on the car. After discussion it was decided to replace with heavier parts the parts suspected of giving trouble. The program we laid out would have required about 3 months. I thought it over that evening, and the next morning proposed to proceed a little differently. So we towed an offending car behind another car and dropped in the clutch. We threw off the fan-belt and took one moving part after another off the engine until nothing was left but the crankshaft, connecting-rods and pistons; still most of the vibration remained. Not having aluminum pistons available, we cut down the heads of those we had until they weighed about 13 oz., the estimated weight of aluminum pistons. At that point most of the vibration ceased. That assured us that lightening the pistons and reducing the reciprocating weight would give us the greatest improvement. This is a good way to check a product.

### Overcoming Resonance

On the same car was an aluminum roof, as beautiful as the side panels of the body. There was considerable vibration in the body and I wished to put on a softer roof that would not produce such a resonating effect in the body. An engineer, in an argument for a change, must sometimes resort to simple fundamental experiments to prove his point. I remembered an old experiment, that we used to perform at school, of sprinkling sand on a plate, vibrating the plate with a violin bow and seeing regular designs form. To prove that the roof was vibrating, we sprinkled sand over the roof and obtained beautiful designs when the engine was running, one design over the front seat and two over the rear seat. Then we put heavy sand-bags on those spots, deadening the vibration. When we showed this to the officials, they said "That's fine; how can we put this weight on the aluminum roof?" I objected, explaining that the roof was a resonator that was causing all the trouble, and that we should use a soft roof. They insisted on retaining that beautiful roof, and we actually put on hundreds of little lead weights, attached by bolts, to stop the vibration, but eventually we adopted a soft roof.

The wheelbase of the small car would probably be about 100 in., possibly a few inches longer. Wheelbases of the other cars would be in proportion. I think the size and weight of the large, luxurious car in America will not be much decreased. The change probably will apply only to small and medium-size cars.

### Narrow Treads

A Member:—In Europe they usually make the tread narrower than it is here. If the wheelbase is 96 in. and the tread 56 in., the car presents a chopped-off appearance. The appearance is not the same if the car is narrow.

Mr. Little:—That is true, but some of these cars must occasionally go on rough roads in the country; and a narrow-tread car is not good on a muddy road.

Mr. Brownback:—In the case I mentioned, the car was very unstable until they made the front tread narrower than the rear tread.

A Member:—Can riding-quality be obtained in the light car as readily as in a heavier car? Will that lead to some departure from the present spring-practice, or will good riding be confined to cars of the better class?

Mr. Little:—I think a decided departure in the design of springs will be necessary. I do not agree with some of the spring manufacturers that a good-riding car can be secured with only a few leaves in the spring.

A Member:—The riding-quality depends upon the proportion of the sprung to the unsprung weight. How can we maintain that?

A Member:—The same proportion of sprung to unsprung weight cannot be obtained in the small car as in a large one.

Mr. Little:—I think we should replace the front brakes with aluminum brakes, thus cutting down the unsprung weight.

A Member:—Do you think the parking problem will eventually have much effect on the wheelbase?

Mr. Little:—Yes; I have often noticed the ease of parking a Ford.

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## CORRECTION IN WINCHESTER PAPER

THROUGH an error in the last paragraph on p. 69 of the paper entitled Operation and Maintenance in the July issue of The Journal, J. F. Winchester's meaning was reversed. The first example given in this paragraph should read as follows:

> We have found the use of single-point boring-tools instead of line reamers for connecting-rod and main bearings results in a more accurate job and requires much less hand work.

# Engine-Acceleration Tests

*By* JOHN O. EISINGER[1]

SEMI-ANNUAL MEETING PAPER *Illustrated with* PHOTOGRAPH AND CHARTS

REPORTING on the progress of the Cooperative Fuel Research, the author considers first the general subject of engine acceleration and then describes the test equipment and procedure of the present investigation. Discussing the degree of accuracy essential and before presenting the results obtained, he calls attention to the determination of a change in engine-torque of the order of 1 lb-ft. if a tachometer or a chronograph is used to determine acceleration. The smallest change in engine-speed that could be detected by the equipment used in this investigation was 2½ r.p.m. per sec., which is equivalent to a torque of about 2 lb-ft.

The calculation of theoretical acceleration is explained. A description of the first series of engine tests, consisting of a comparison of the maximum acceleration obtained with three different carbureters used without accelerating wells, is included. A series of tests was made, using one of these carbureters, to ascertain the effect of changing jacket-water temperature alone. A somewhat similar set of tests was made in an effort to determine the effect of intake-manifold temperature, and the results are presented. The effect on performance of injecting various amounts of accelerating charge was determined also for each of the carbureter adjustments. Engine tests made recently in an attempt to determine the cause of the leaning of the mixture when the throttle is opened suddenly are analyzed, as well as the subject of flow of liquid fuel along the intake-manifold wall. Methods of determining the moment of inertia are presented in the Appendix.

It is stated in the discussion following the paper that the velocity of the fuel mixture is a very important factor, and some experiments are cited that seem to corroborate the statement. Other subjects discussed include the apparent lag and subsequent recovery indicated by the charts, the variations in the duplicate runs, and the difficulty of obtaining consistent results when trying to measure engine acceleration.

ONE of the major problems with which the Cooperative Fuel Research[2] has been concerned for the last 6 months has been a study of engine acceleration. This investigation was undertaken in accordance with a recommendation made by the Steering Committee on Dec. 4, 1925, which states that

> The study of engine starting which has thus far dealt with the obtaining of the initial start ultimately should cover the behavior of the engine during the "warming-up" period, particularly as regards *acceleration.*

Before attempting to study engine acceleration during the warming-up period, it seemed desirable to consider the general subject of engine acceleration. Motor-car acceleration or flexibility depends, not only on the ability of the engine to accelerate, but also on such matters of car design as the gear reduction from the engine to the rear wheels and the ease of gear-shifting. The present investigation has reference to engine rather than to car acceleration and, in particular, to the effects of engine characteristics, fuels and carbureters on engine acceleration.

During the 5 years of its history, the Cooperative Fuel Research has been concerned with various problems and has at times undertaken a study of engine acceleration in connection with the general problems then at hand. The first work on acceleration was done in 1923 in conjunction with the economic-motor-fuel survey. These acceleration tests were made with two makes of car and two grades of gasoline. In discussing the limited amount of data obtained it was stated[3] that "the results of these tests are erratic, probably due to the wide variation in engine temperature and due to wind." However, the general conclusion reached, that no consistent and significant differences could be detected between the test fuels in fuel economy or any other phase of performance, was substantiated later by laboratory tests. These supplementary laboratory tests, where engine conditions could be controlled more easily and accurately, showed[4] further that "the maximum acceleration occurs at the carbureter setting giving maximum power at 600 r.p.m. and full throttle." It was with this background that the present investigation was undertaken.

## TEST EQUIPMENT AND PROCEDURE

The first tests were made on the four-cylinder motor-truck engine that was used in the engine-starting tests. As a result of these preliminary tests, a tentative method of procedure was developed. The Steering Committee, however, recommended that the main series of tests be conducted on a passenger-car engine of modern design. Therefore, a 1926-model 3⅛ x 4½-in. six-cylinder engine having a piston displacement of 207 cu. in. was set up as shown in Fig. 1 and the necessary equipment was installed in accordance with previous experience.

To make it certain that the engine in the laboratory would have an acceleration similar to that which it would have if used to propel a car, the following conditions were imposed on the engine: First, the inertia due to the weight of the car was simulated by the usual laboratory method, which consists in coupling to the end of the dynamometer shaft a steel disc or flywheel[5], the inertia of which, together with that of the dynamometer armature, is equivalent to that of the car. Before the size of the inertia disc could be calculated, the

[1] S.M.S.A.E.—Research associate, Bureau of Standards, City of Washington.

[2] This Cooperative Fuel Research has been in progress at the Bureau of Standards since August, 1922, under the direction of a Steering Committee composed of representatives of the cooperating organizations, which are the American Petroleum Institute, the National Automobile Chamber of Commerce, the Bureau of Standards and the Society. For reports of the previous work, see bibliography at the end of the paper.

[3] See THE JOURNAL, July, 1923, p. 5.

[4] See THE JOURNAL, March, 1924, p. 270.

[5] See THE JOURNAL, March, 1924, p. 271.

moment of inertia of the dynamometer armature had to be determined. The methods used in making this determination are described in the Appendix to this paper. Second, to impose on the engine a load similar to the wind and rolling resistance that a car encounters on the road, the engine was loaded by the dynamometer. The load normally used is shown in Fig. 2.

The general test-procedure was as follows: With a given carbureter adjustment and operating temperatures, the engine usually was adjusted to idle at 250 r.p.m. By a special arrangement of electromagnets which held the throttle in the idling position and an electrical contact driven at camshaft speed, the throttle was opened automatically at a definite point in the cycle of cylinder No. 1. This assures the beginning of each acceleration run under the same flow-conditions in the intake-manifold. The exact length of the idling time before each run, as brought out in preliminary tests on this engine, was not important for most of the conditions covered in this report, but the idling time was usually about 2 min. The spark was maintained in the fully advanced position, since this gave the best results when accelerating under normal load from closed-throttle position. The throttle was instantly opened automatically and, at the end of each second, the average engine-speed during that second was indicated by the position of the hand of a special chronometric tachometer. The time of opening the throttle and the readings of the tachometer were synchronized by head telephones.

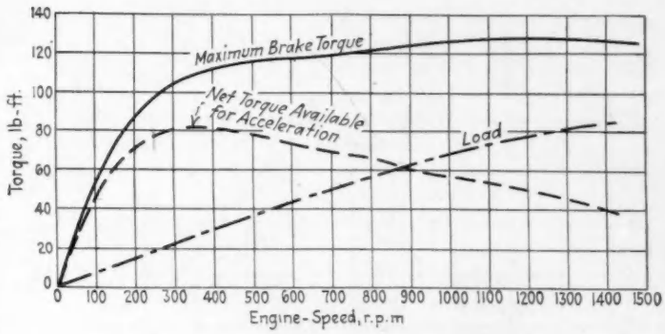


FIG. 2—CHARACTERISTIC CURVES

The Curves Show, Respectively, the Maximum Brake-Torque of the Engine, the Load Imposed Normally on the Engine and the Net Torque Available for Acceleration. The Jacket-Water Temperature Was 70 Deg. Cent. (158 Deg. Fahr.) and the Temperature of the Air Was 28 Deg. Cent. (82 Deg. Fahr.)

The position of the hand of the tachometer was recorded manually every second on a paper scale attached to the face of the tachometer. At the conclusion of a run these indicated readings were entered on the data sheets before a second run was started. The tachometer used reads speed in multiples of 2½ r.p.m.; so the possible error in any reading was ± 1¼ r.p.m. Two burettes,

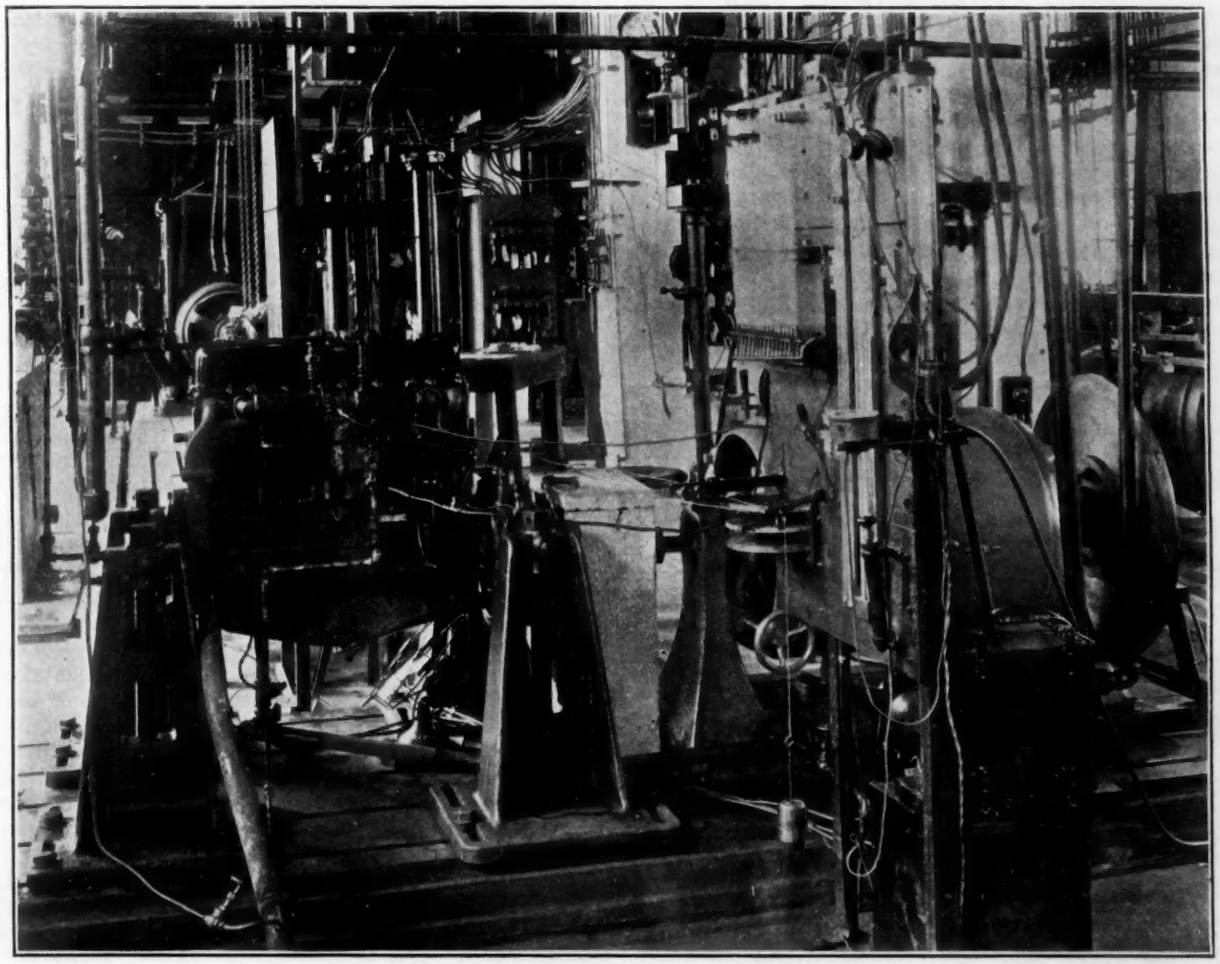

FIG. 1—ENGINE-ACCELERATION-TEST EQUIPMENT

The Engine Shown Is a 1926-Model, 3⅛ x 4½-In., Six-Cylinder Type Having a Piston Displacement of 207 Cu. In.

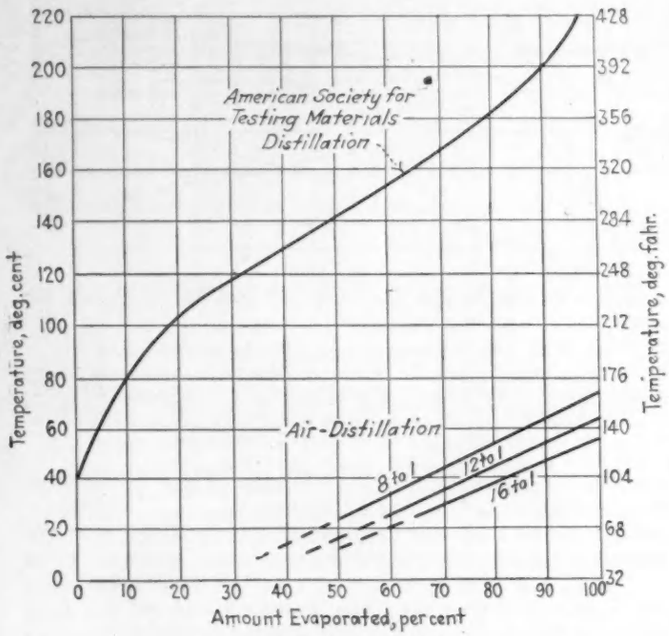


FIG. 3—DISTILLATION CURVES

The Gasoline Used in Most of the Investigation Was Purchased on Government Specifications and Is Designated as Commercial Gasoline No. 11. Its Distillation According to the Requirements of the American Society for Testing Materials, As Well As Its Air Distillation, Is Shown

each of 50-cc. capacity, were connected to the carbureter and supplied the gasoline used. The fuel consumed during an acceleration run could be determined by marking on a paper scale the position of the meniscus of the gasoline in the burette at the start and at the end of the run. The carbureter adjustment was identified by a full-throttle run at 1000 r.p.m. The gasoline used in most of this investigation was purchased on Government specifications, and its distillation according to the requirements of the American Society for Testing Materials, as well as its air distillation, is shown in Fig. 3.

Before presenting the results obtained on the engine, attention is called to the accuracy necessary to determine a change in engine-torque of the order of 1 lb-ft., if a tachometer or a chronograph is used to determine acceleration. With the inertia of 8.61 lb. sec.$^2$ ft., explained in the Appendix to this paper, which the engine under test has to overcome, a torque of 1 lb-ft. when acting for 1 sec. will produce a change in speed of:

$$\alpha = T/I \qquad (1)$$

where

$\alpha$ = angular acceleration of the engine in radians per second per second
$I$ = rotational moment of inertia
$T$ = torque in pound-feet

Substituting values in equation (1), we have

$$\alpha = 1/8.61 \qquad (2)$$
$$= 0.116 \text{ radians per sec. per sec.}$$

But $2\pi$ radians is the equivalent of 1 revolution, and 0.116 radians per sec. per sec. equals 0.0185 revolutions per sec. per sec. Therefore, the angular velocity at the end of 1 sec. will be 0.0185 revolutions per sec., or 1.11 r.p.m.; consequently, at any engine-speed, it will be necessary to determine a change in speed of about 1 r.p.m. per sec. The smallest change in engine-speed that could be detected by the equipment used in this investigation was 2½ r.p.m. per sec., which is equivalent to a torque of about 2 lb-ft.

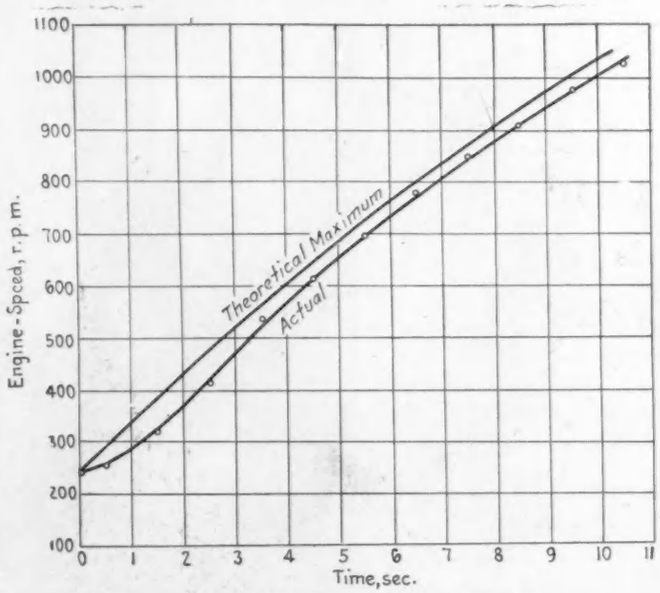


FIG. 4—THEORETICAL AND ACTUAL SPEED-TIME CURVES

It Is Evident That the Agreement between the Actual Engine-Test Curve and the Theoretical Maximum Curve as Calculated Is Good. The Respective Temperatures Were Air, 24 Deg. Cent. (75 Deg. Fahr.); Jacket-Water, 79 Deg. Cent. (174 Deg. Fahr.); and Intake-Manifold, 85 Deg. Cent. (185 Deg. Fahr.)

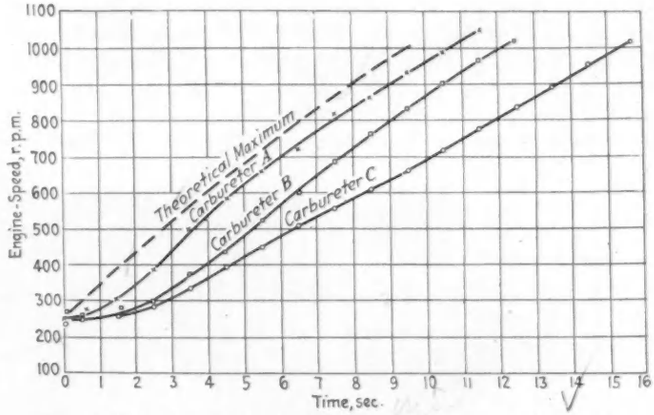


FIG. 5—MAXIMUM ACCELERATION OBTAINED WITH DIFFERENT CARBURETERS

Carbureter *A* Is a Plain-Tube Type. Carbureters *B* and *C* Are of the Air-Valve Type. The Respective Temperatures Were Air, 26 Deg. Cent. (79 Deg. Fahr.); Jacket-Water, 85 Deg. Cent. (185 Deg. Fahr.); and Intake-Manifold, 97 Deg. Cent. (207 Deg. Fahr.) Data Relating to the Tests, Which Were Made at an Engine-Speed of 1000 R.P.M., Are Given in the Following Tabulation

| Carbureter | Total Fuel Used per Acceleration Run, Cc. | Torque, Lb-Ft. | Air-Fuel Ratio |
|---|---|---|---|
| *A* | 18.4 | 118 | 13.7 |
| *B* | 34.6 | 113 | 11.1 |
| C | 16.0 | 115 | 13.2 |

Increased accuracy could be obtained by using a chronograph, but this would involve experimental difficulties at the higher engine-speeds. The use of experimental speed-time curves to obtain acceleration is not ideal, since the quantity sought is the time derivative of such curves. Efforts are therefore being directed to the development of an instrument capable of indicating directly the engine-torque under accelerating conditions.

## CALCULATION OF THEORETICAL ACCELERATION

Since torque and acceleration are directly proportional, the theoretical acceleration can be calculated for this engine if the torque at the various speeds is known. With the aid of acceleration values computed from the net torque shown in Fig. 2, a maximum speed-time curve was constructed for comparison with the test results as

shown in Fig. 4. The extent to which test curves deviate from this maximum curve will indicate in a general way the deficiency of engine-torque when accelerating which is chargeable to fuel, carbureter or intake system. As a check on this theoretical curve, an engine test was made in which the acceleration was started while the engine was turning over at 250 r.p.m. with the throttle fully opened; that is, the engine was loaded down to 250 r.p.m. with the throttle wide-open and, at the beginning of the run, the load was decreased quickly to that corresponding to the load normally employed. This procedure eliminates only the lag in acceleration resulting from a sudden drop in manifold depression, and there may still be some lag in providing the greater volume of charge required for increased engine-speed. However, the agreement between the actual engine-test and the calculated curve is surprisingly good, as is shown in Fig. 4. For purposes of comparison, the theoretical curve is represented by a dotted line in Figs. 5 to 11 inclusive.

### First Series of Tests

The first series of engine tests consisted of a comparison of the maximum acceleration obtained with three different carbureters, used without accelerating wells, the results being shown in Fig. 5. Carbureter *A* is a plain-tube type; carbureters *B* and *C* are of the air-valve

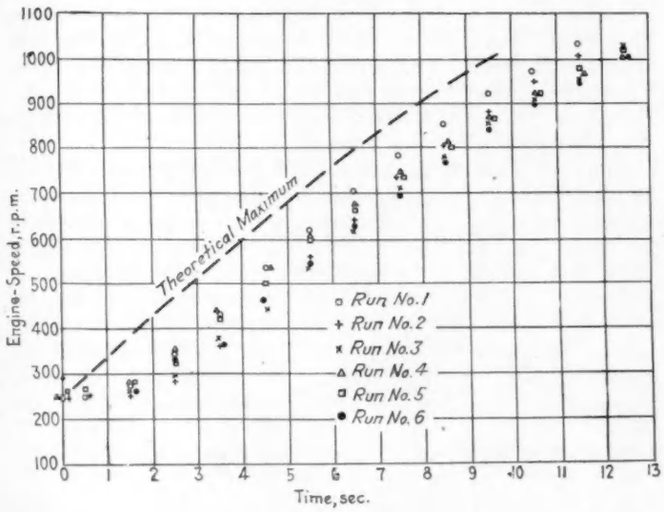


Fig. 6—Effect of Jacket-Water Temperature upon Acceleration

Data for the Various Runs, Taken at an Air Temperature of 26 Deg. Cent. (79 Deg. Fahr.), and an Engine-Speed of 1000 R.P.M., Are Given in the Following Tabulation

| Run No. | Temperatures: Intake-Manifold Deg. Cent. | Intake-Manifold Deg. Fahr. | Jacket-Water Deg. Cent. | Jacket-Water Deg. Fahr. | Total Fuel Used per Acceleration Run, Cc. | Torque, Lb-Ft. | Air-Fuel Ratio |
|---|---|---|---|---|---|---|---|
| 1 | 81 | 178 | 66 | 151 | 19.0 | 119 | 13.3 |
| 2 | 88 | 190 | 82 | 180 | 18.8 | ... | ... |
| 3 | 95 | 203 | 100 | 212 | 17.6 | 109 | 17.2 |
| 4 | 77 | 171 | 52 | 126 | 21.2 | 120 | 12.9 |
| 5 | 73 | 163 | 36 | 97 | 25.8 | 121 | 12.3 |
| 6 | 69 | 156 | 12 | 54 | 23.8 | 118 | 12.1 |

type. These acceleration tests, and most of those which follow, cover a speed range from 250 to 1000 r.p.m. and it will be noted that the most important and pronounced lag in acceleration always occurs during the time the engine-speed is increasing from 250 to 350 or 450 r.p.m. Fig. 5 shows that different carbureters may give marked differences in performance under normal temperature-conditions. Because of its better performance shown here and the more consistent results obtained with it in other tests, carbureter *A* was used in all subsequent tests. The accelerating well was permanently disconnected and was replaced by a special device for injecting an accelerating charge whenever desired. Preliminary tests showed that any setting of carbureter *A* supplied practically the same mixture-ratio throughout the speed range covered in these tests.

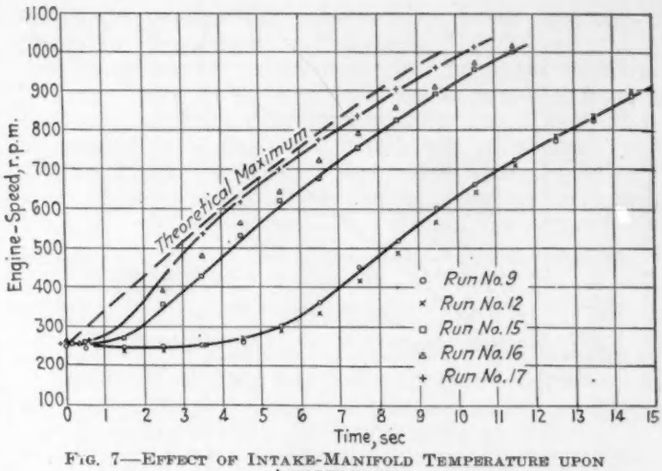


Fig. 7—Effect of Intake-Manifold Temperature upon Acceleration

Data for the Various Runs, Taken at an Air Temperature of 25 Deg. Cent. (77 Deg. Fahr.), and an Engine-Speed of 1000 R.P.M., Are Given in the Following Tabulation

| Run No. | Temperatures: Intake-Manifold Deg. Cent. | Intake-Manifold Deg. Fahr. | Jacket-Water Deg. Cent. | Jacket-Water Deg. Fahr. | Total Fuel Used per Acceleration Run, Cc. | Torque, Lb-Ft. | Air-Fuel Ratio |
|---|---|---|---|---|---|---|---|
| 8 | 32 | 90 | 36 | 97 | Engine would not accelerate | | |
| 9 | 42 | 108 | 66 | 151 | 30.6 | 123 | 11.6 |
| 12 | 71 | 160 | 92 | 198 | 29.6 | 119 | 13.3 |
| 15 | 78 | 172 | 96 | 205 | 23.6 | 118 | 13.7 |
| 16 | 95 | 203 | 95 | 203 | 21.2 | 114 | 14.2 |
| 17 | 87 | 189 | 94 | 201 | 22.0 | 116 | 13.7 |

The influence of jacket-water temperature on acceleration is shown in Fig. 6. In these tests jacket-water temperature alone was changed, all other temperatures, such as oil temperature and intake-air temperature, being kept as nearly constant as possible. While a slight enrichment of the mixture was necessary to produce the same performance at cold jacket-water temperature as was obtained at higher temperatures, the enrichment required is fairly small considering the marked change in engine-cylinder temperature. It seems that the temperature of the cylinder-walls has very little effect on the acceleration obtained.

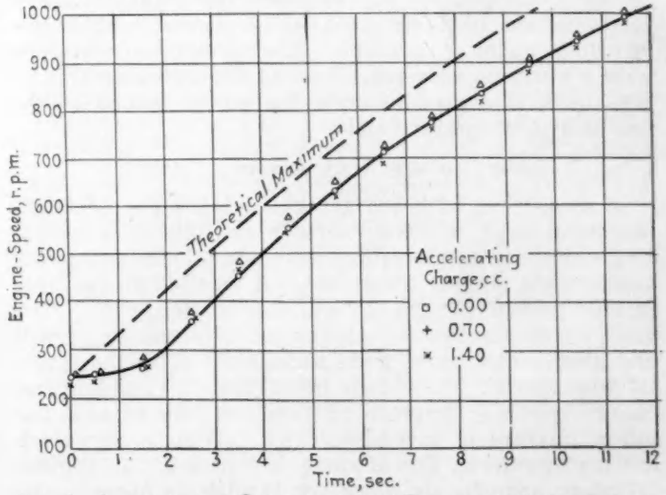


Fig. 8—Acceleration Obtained with Different Quantities of Accelerating Charge

The Engine-Speed Was 1000 R.P.M.; the Air-Fuel Ratio, 9.2; and the Torque, 111 Lb-Ft. The Respective Temperatures Were Air, 26 Deg. Cent. (79 Deg. Fahr.); Jacket-Water, 86 Deg. Cent. (187 Deg. Fahr.); and Intake-Manifold, 84 Deg. Cent. (183 Deg. Fahr.)

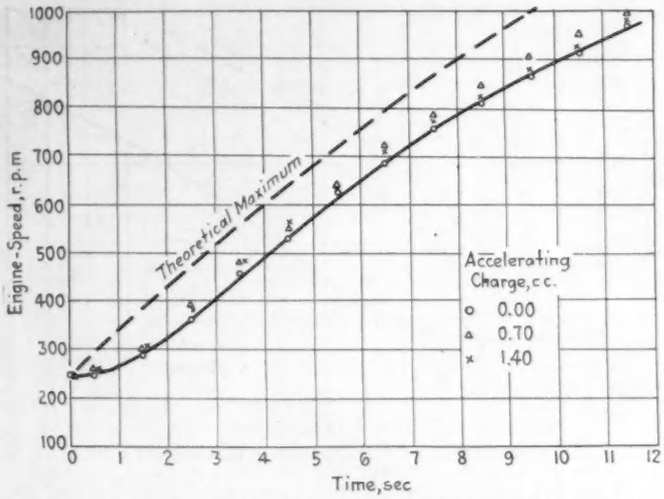


FIG. 9—ACCELERATION OBTAINED WITH DIFFERENT QUANTITIES OF ACCELERATING CHARGE

The Engine-Speed Was 1000 R.P.M.; the Air-Fuel Ratio, 11.4; and the Torque, 113 Lb-Ft. The Respective Temperatures Were Air, 26 Deg. Cent. (79 Deg. Fahr.); Jacket-Water, 86 Deg. Cent. (187 Deg. Fahr.); and Intake-Manifold, 84 Deg. Cent. (183 Deg. Fahr.)

A somewhat similar set of tests was made in an effort to determine the effect of intake-manifold temperature on acceleration. The intake-manifold is jacketed completely and its temperature is controlled by varying the amount of steam entering this jacket. The results are shown in Fig. 7. The acceleration obtained under a given set of conditions is largely dependent upon the temperature of the intake-manifold[6]. With the particular gasoline used, an intake-manifold temperature below about 78 deg. cent. (172 deg. fahr.) causes a decided lag in the acceleration of the engine. Although the intake-manifold temperature of 71 deg. cent. (160 deg. fahr.) in run No. 12 is well above the equilibrium-air-distillation temperatures for complete evaporation at a mixture-ratio of 13.3 to 1.0, as shown in Fig. 3, the unsatisfactory performance of the engine in this run seems to indicate lack of complete vaporization under these operating conditions.

The effect of mixture-ratio on acceleration is shown in Figs. 8 to 11 inclusive. The least lag in acceleration naturally occurs when operating with rich mixtures and, as has been pointed out by other investigators, such mixtures are likely to be used in service with corresponding waste of gasoline. The acceleration obtained with a given mixture-ratio, beyond the initial portion of the curve, is dependent upon the power output of the engine at that mixture-ratio.

### INJECTION OF ACCELERATING CHARGE

In connection with the foregoing tests, the effect on performance of injecting various amounts of accelerating charge was determined for each of the carbureter adjustments already mentioned. A small-dash gasoline-primer served in lieu of an accelerating well. The quantity of accelerating charge could be varied at will and always was introduced immediately after the throttle was opened. It will be noted that the accelerating charge ceases to improve acceleration only in case the idling mixture is excessively rich. With a very rich carbureter-setting, the mixture delivered to the engine-cylinders immediately after the throttle is opened may not become lean enough to affect acceleration materially, but such an adjustment gives very high average fuel-consumption.

A few tests were made to determine the effect of a change in the location of the accelerating jet. The accelerating jet was placed at three different points in the intake system: (*a*) at the carbureter-air entrance, (*b*) just above the throttle and (*c*) at the top of the vertical portion of the intake-manifold. The accelerating charge was always discharged in the direction of the airstream. No consistent or significant difference attributable to the location of the jet could be detected when using the same quantity of accelerating charge.

### EFFECT OF SUDDEN THROTTLE-OPENING

Precisely what happens when the throttle is opened suddenly and what the cause is of the leaning of the mixture have been discussed frequently. Various theories have been suggested. Some engine tests have been made recently in an attempt to determine the cause of this phenomenon.

One of the possible causes may be the inertia of the gasoline in the carbureter jets and differences in the coefficient of discharge of the jets under accelerating conditions. That the lag in the flow of liquid from the carbureter jets is small, at least in carbureter *A*, was demonstrated by a variable-orifice flow-meter sensitive to a change in fuel-flow of 0.025 cc. per sec. This instrument showed immediate response to the full-throttle flow-rate, about 0.8 cc. per sec., when the throttle was opened suddenly. Therefore, the lag in discharge of fuel from the carbureter jets is extremely small when compared to the lag of 2 and 3 sec. that frequently occurs in the acceleration of the engine.

Another possible reason for the leaning of the mixture is that, whereas the fuel is discharged from the carbureter jets into the airstream, a very considerable flow of liquid fuel takes place along the intake-manifold walls. This flow is said to be extremely slow, and the quantity entering the liquid film is dependent upon the intake velocity. That this is a contributing cause is brought out in the acceleration runs made with lean mixtures, as shown in Fig. 11. Reference is made also to a paper on Fuel-Supply Requirements for Acceleration[7], by P. S. Tice.

The runs made when using the smaller quantities of

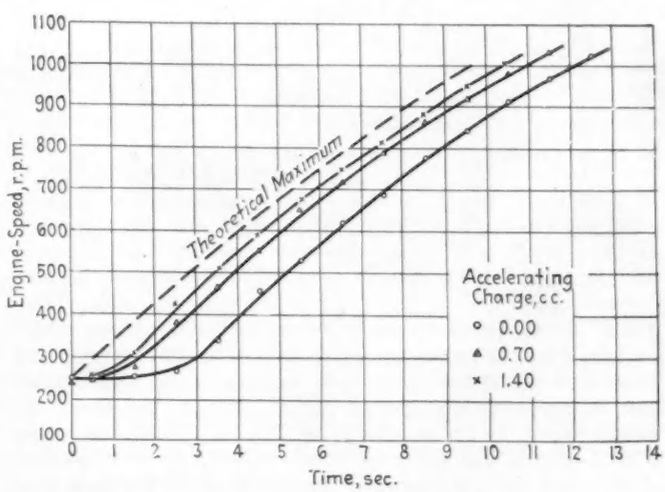


FIG. 10—ACCELERATION OBTAINED WITH DIFFERENT QUANTITIES OF ACCELERATING CHARGE

The Engine-Speed Was 1000 R.P.M.; the Air-Fuel Ratio, 15.3; and the Torque, 116 Lb-Ft. The Respective Temperatures Were Air, 23 Deg. Cent. (73 Deg. Fahr.); Jacket-Water, 86 Deg. Cent. (187 Deg. Fahr.); and Intake-Manifold, 85 Deg. Cent. (185 Deg. Fahr.)

[6] See THE JOURNAL, August, 1920, p. 131.

[7] See *Automotive Industries*, July 9, 1925, p. 53.

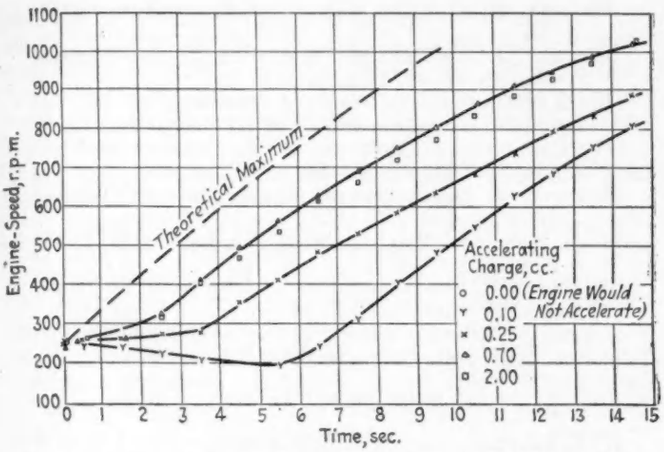


FIG. 11—ACCELERATION OBTAINED WITH DIFFERENT QUANTITIES OF ACCELERATING CHARGE

The Engine-Speed Was 1000 R.P.M.; the Air-Fuel Ratio, 16.5; and the Torque, 102 Lb-Ft. The Respective Temperatures Were Air, 23 Deg. Cent. (73 Deg. Fahr.); Jacket-Water, 86 Deg. Cent. (187 Deg. Fahr.); and Intake-Manifold, 86 Deg. Cent. (187 Deg. Fahr.)

accelerating charge show a decided lag in acceleration which extends over a period of 3 to 5 sec. During this interval, gasoline is being delivered by the carbureter in quantities proportional to the increase in airflow; but, apparently, the increase in fuel does not reach the engine-cylinder until several seconds have passed. For a few seconds following this period of lag, the mixture delivered to the engine-cylinder is much richer than the mixture for which the carbureter is adjusted. Therefore, when operating with a lean mixture, the best acceleration is obtained shortly after the period of lag.

### FLOW OF LIQUID IN THE INTAKE-MANIFOLD

In connection with the flow of liquid fuel along the intake-manifold wall, the results shown in Fig. 12 are of particular interest because they indicate the performance obtained when accelerating from various engine-speeds without injecting any accelerating charge. Accelerating from 250 r.p.m. produces a decided lag in the acceleration. Starting from a speed 12 r.p.m. higher, or at 262 r.p.m., a great improvement in acceleration is obtained. With further increases in the starting-speed of 10 and 16 r.p.m., similar improvement results. However, increasing the initial speed from 288 to 325 r.p.m. gives less improvement in performance than was caused by a change of 10 to 16 r.p.m. at speeds below 288 r.p.m. These comparisons were made with a lean carbureter-adjustment so that any differences which existed would be intensified. If the lag in acceleration were due entirely to the lag in the flow of liquid fuel along the intake-manifold wall, it seems that a gradual and uniform improvement in acceleration should be obtained with equal increase in the starting-speed.

The fact that the improvement is not uniform may be due to various causes. There may be (*a*) marked changes in the character of the pulsations in the intake-manifold at certain speeds and (*b*) a critical speed below which the manifold velocity is insufficient to carry all the liquid fuel, especially to raise it through the vertical portion of the manifold. That fuel falls out of the incoming charge at the low engine-speeds is confirmed on comparing the slopes of the various curves at speeds above 500 r.p.m. Within certain limits, the longer the period of no acceleration is, the greater is the quantity of liquid accumulating in the intake-manifold and, hence, the greater and more prolonged will be the improvement in acceleration thereafter.

In conclusion, the work done thus far gives some information as to the relative influence of speed-range, carbureter characteristics, jacket-water temperature, intake-manifold temperature, carbureter adjustment, and the injection of an accelerating charge upon the acceleration performance of a given engine using the same fuel. The next step seems to be a study of the acceleration performance obtainable with different fuels under a variety of temperature conditions.

## APPENDIX

### MOMENT-OF-INERTIA DETERMINATIONS

The rotating mass to be coupled to the engine so as to produce the same effect as that due to the weight of the car was determined by the following procedure.

The engine under test is used normally in a car which, when loaded with two passengers, would have a weight of about 3400 lb. The rear-axle ratio is 4.9 and the tires used are about 31 in. in diameter. The rotational inertia-equivalent to the weight of the car is given by equation (7) of the paper by Roger Birdsell on Economic Motor-Fuel Volatility[5], as follows:

$$I = (1/4)\ (W/g)\ (D^2/R^2) \qquad (3)$$

where

$D$ = diameter of rear wheel in feet
$g$ = acceleration of gravity in feet per second per second
$I$ = moment of inertia
$R$ = rear-axle ratio
$W$ = weight of car in pounds

Substituting values in equation (3) we have

$$I = (1/4)\ (3400/32.2)\ [(31)^2/(12)^2\ (4.9)^2]$$
$$= 7.34 \text{ lb. sec.}^2 \text{ ft.} \qquad (4)$$

Hence, the rotating mass to be coupled to the engine should have a moment of inertia of 7.34 lb. sec.$^2$ ft. The next step consists in the determination of the moment of inertia of the dynamometer armature so that, if its magnitude is less than that required, the additional inertia necessary can be determined.

One of the methods used to determine the moment of inertia of the dynamometer armature may be referred to as a constant-torque method. With the dynamometer uncoupled from the engine, a constant electrical torque, the value of which is observed on the dynamometer scale, is applied and the armature accelerates. From the speed-time curve obtained from the observations made

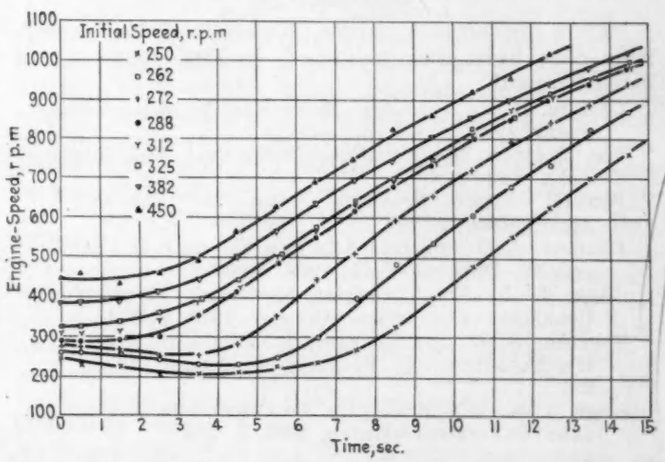


FIG. 12—RESULTS OBTAINED WHEN ACCELERATING FROM DIFFERENT ENGINE-SPEEDS

The Respective Temperatures Were Air, 31 Deg. Cent. (88 Deg. Fahr.); Jacket-Water, 84 Deg. Cent. (183 Deg. Fahr.); and Intake-Manifold, 88 Deg. Cent. (190 Deg. Fahr.)

[5] See THE JOURNAL, March, 1924, p. 272.

on a chronometric tachometer, which is attached to the dynamometer, the angular acceleration of the armature can be determined. The moment of inertia about the axis of rotation can be computed from the equation:

$$I = T/\alpha \quad (5)$$

where

$\alpha$ = angular acceleration in radians per second per second
$\Delta v$ = change in engine-speed in revolutions per minute per second
$T$ = torque in pound-feet

But

$$\alpha = (2\pi/60)\ \Delta v \quad (6)$$
$$= \pi\Delta v/30 \quad (7)$$

Therefore

$$I = 30\,T/\pi\Delta v \quad (8)$$

Applying a torque of 3.28 lb-ft., the change in engine-speed of $\Delta v = 27$ r.p.m. per sec. was observed. Therefore, from equation (8),

$$I = (30 \times 3.28)/(\pi \times 27) \quad (9)$$
$$= 1.16 \text{ lb. sec.}^2 \text{ ft.} \quad (10)$$

The moment of inertia of the dynamometer armature being equal to 1.16 lb. sec.$^2$ ft. requires that an additional mass having an inertia of $7.34 - 1.16 = 6.18$ lb. sec.$^2$ ft. be coupled to the dynamometer armature so as to approximate the inertia of the car. A steel disc was designed and its moment of inertia was determined later by two different methods; (*a*) from the direct measurements of its size and weight and (*b*) by a torsion-pendulum experiment. The value obtained by method (*a*) was 6.18 lb. sec.$^2$ ft. and by method (*b*) 6.17 lb. sec.$^2$ ft.

Since the inertia of the moving parts of the engine is also a factor in any acceleration tests, an experiment was made to determine the magnitude of this inertia. By a constant-torque experiment the moment of inertia of the moving parts of the engine was found to be 1.28 lb. sec.$^2$ ft. When traveling in high gear this is approximately 17 per cent of the total inertia due to the weight of the car.

The engine, during acceleration, must overcome the inertia of the moving parts of the engine and also the inertia of the dynamometer armature and the acceleration disc. The total of these three items is equal to 8.61 lb. sec.$^2$ ft. and is the moment of inertia used in calculating the maximum theoretical acceleration referred to at the beginning of this paper.

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### THE DISCUSSION

W. G. CLARK[9]:—Is my understanding correct that Mr. Eisinger's experiments showed the velocity of the fuel mixture to be a very important factor?

J. O. EISINGER:—Yes.

MR. CLARK:—Some experiments we made several years ago seem to corroborate that statement. In working on kerosene carburetion, we found that velocity is a very important factor in maintaining the mixture from the carbureter to the cylinder. In line with that investigation, we made some experimental manifolds with the cross-section reduced to increase the velocity of flow. The acceleration and the running of the engine were improved materially. But we encountered volumetric limits that reduced the maximum power obtainable, and we could increase the velocity to a limited extent only.

We attempted first to use a cast manifold; that is, a multiple manifold similar to a group of pipes. Since the friction losses due to the roughness of the castings were high enough to prevent getting satisfactory results, we made a manifold by hand. We used copper tubes, of ¼-in. outside diameter and 3/16-in. inside diameter, soldering them together. A throttle was inserted where the two sections of the manifold joined. It had the form of a piston which uncovered layers of tubes as it was moved. In this manner, the effect of a constant velocity at all throttle openings was obtained by uncovering a certain number of tubes for each given engine-speed. In other words, the mixture was allowed to flow through just enough of the tubes to supply the engine with sufficient fuel for any given engine-speed. Thus, we obtained remarkably good acceleration and there was no liquid in the tubes so far as we could determine. We could observe no indication of liquid clinging to the walls, due to the high initial velocity and to the smoothness of the tubes. We obtained high volumetric-efficiency by making the total area of the tubes equal to the required amount to give maximum power when all the tubes were uncovered by the throttle. By causing the mixture to flow through only a small number of tubes at low speeds of the engine, the same velocity was attained as would have occurred at high engine-speeds with the mixture flowing through all the tubes as if they constituted a single-pipe manifold. This seems to indicate that velocity is one of the most important considerations in maintaining a dry mixture and maximum accelerating qualities with any given fuel, which corroborates Mr. Eisinger's findings to some extent.

R. V. HUTCHINSON[10]:—Regarding the apparent lag and subsequent recovery indicated by the charts in Mr. Eisinger's paper, this is intimately connected with the ability of an airstream to carry in suspension a series of droplets of liquid fuel. The ability will vary with the

[9] M.S.A.E.—Chief automotive and lubricating engineer, Pure Oil Co., Chicago.

[10] M.S.A.E.—Designing engineer, Olds Motor Works, Lansing, Mich.

density of the airstream and as the square of the velocity. Prof. George B. Upton treats the subject thoroughly in an article entitled Raindrops and Engineering[11]. As to the variation in the action of the mixture with jacket-water temperature, W. A. Geise and I experimented with a Scripps marine engine and found that the operation of an engine of this type is almost independent of jacket-water temperature.

It is possible to obtain a mixture that is dry near the manifold-walls and wet in the center, or vice versa. We have made engine tests, using volatile gasoline, in which the temperature indicated by a thermocouple inserted in the mixture-stream was 100 deg. fahr. higher than that of the Engler end-point of the fuel, and still we found liquid on the walls. However, if the liquid is impinged at very high velocity on a sufficiently hot wall, it will splash off in small wet globules of fairly low temperature, and these globules are perfectly capable of being carried in the mixture-stream. This type of preparation results in dry edges and a wet center of the mixture-stream.

So-called condensation does not occur; because, in many instances, evaporation is never complete due to lack of time. But the collecting of a number of small drops to form a large drop does occur often and, under certain conditions, this drop will fall out, wet the wall and be available for use later during the acceleration period.

MR. CLARK:—The point made by Mr. Hutchinson regarding the condition in which the mixture is wet in the center and dry along the walls was exemplified in the copper-tube manifold I have described. Each tube was a miniature manifold of such small diameter that its center was very close to the wall; so the maximum velocity of the mixture stream was almost next to the wall and this fact prevented the liquid from remaining on the wall. The condition is really that of stratified layers of mixture traveling at different velocities which vary from the maximum, perhaps in the center of the manifold or along the line of least friction, to the minimum, along the wall where the frictional resistance of the wall retards the velocity. If we could make a manifold of very small cross-section and still not have a volumetric loss, the difficulty mentioned by Mr. Hutchinson would be obviated.

MR. HUTCHINSON:—I disagree with Mr. Clark as to the advantages of the multitubular manifold he describes because, in making a multitubular manifold, the ratio of wall surface to cross-sectional area is increased greatly. Further, liquid fuel actually wets metals and tends to adhere to them by virtue of the surface tension existing between them. The consequence is that, once the liquid gets onto the wall and wets it, it is impossible for the liquid to move along that wall with the same velocity as that of the air that is passing. The multitubular manifold mentioned by Mr. Clark only accentuates the wetness; whereas, if the liquid fuel is impinged at very high velocity on a sufficiently hot surface, as I have already described, the liquid fuel is thereby put into such a condition that it can be carried in suspension and not readily thrown out under the influence of its own inertia.

MR. CLARK:—Undoubtedly Mr. Hutchinson is right about the increased friction of the wall surface in a multitubular manifold. We found the same difficulty that he refers to when we tried to cast a multitubular manifold. But by using smooth-drawn copper-tubes, the interior wall surface of which was very smooth, we found no difficulty due to having liquid remain on the inner wall. Because the maximum velocity of the airstream in the small tube was so near the wall, it apparently was sufficient to carry with it any liquid which tended to cling to the wall.

In spite of the apparent scientific discrepancies in the multitubular manifold I described, the fact that it worked is the main consideration; and we were able in this manner to carburet a good grade of kerosene successfully without the addition of any external heat.

DR. J. C. GENIESSE[12]:—The charts in Mr. Eisinger's paper lead one to believe that each curve was drawn through one series of values. If that is true, or if the points through which the curves are drawn are averages, the paper should specify which is the case. Also the paper should give the variations obtained with duplicate runs, to give the reader an opportunity to evaluate the data correctly.

MR. EISINGER:—I am glad that Dr. Geniesse has brought up these points. Apparently, he realizes the difficulty that is encountered in obtaining consistent results when trying to measure engine acceleration. We devoted considerable time to developing a satisfactory method of measuring engine acceleration and to the modification of the engine so as to obtain consistent results.

Regarding the points for the curves, the points as shown in the charts are observed values taken from a representative run.

As to the variations in duplicate runs, a check run was always made under a given set of conditions and the differences in the final engine-speed for any two runs, when accelerating for the same length of time and when operating with normal mixtures, was seldom greater than 30 r.p.m. and in most cases considerably less. However, for the lean-mixture runs, larger differences would be expected and, under such conditions, it was necessary to make a larger number of check runs.

F. C. MOCK[13]:—Mr. Eisinger's report is of extreme value, it seems to me, in that it bears upon what is economically the most important phase of carburetion as regards passenger-cars. We make careful dynamometer measurements of the fuel required for maximum power, for steady running at different speeds and loads, and take long continuous drives to determine tank-mileage. But if we watch the procedure at the different garages and service stations we find that carbureters are seldom adjusted with a particular view to speed, hill climbing or economical steady running. It seems to be a universal custom to adjust the carbureter according to the acceleration, by "flashing" the throttle open quickly and opening-up the fuel feed as may be necessary to give the best response; that is, carbureters in daily service are adjusted solely on the basis of acceleration.

The tests made by Mr. Eisinger show conclusively that, through a large range of temperatures, prompt acceleration can be obtained with considerably less fuel-consumption by using an "accelerating charge" and a lean mixture for steady running than by using the richer steady-running mixture necessary when no accelerating charge is used. It is significant that the American cars which do not use a supplementary accelerating-charge fall into two classes; (*a*) those having a low selling-price which renders first cost paramount over efficient operation and (*b*) those having an intake-manifold-heating system that was designed by the designer

[11] See *Sibley Journal of Engineering*, January, 1923, p. 2; February, 1923, p. 32; and March, 1923, p. 60.

[12] Atlantic Refining Co., Philadelphia.

[13] M.S.A.E.—Research engineer, Stromberg Motor Devices Co., Chicago.

of the carbureter and in which the heat application to the intake charge is unusually great.

As reported, Mr. Eisinger's experiments apparently did not reach a point at which 1 cc. of fuel accelerating-charge was too great. On a fairly large number of cars we have found that, after cross-country driving in warm weather, even a 0.5-cc. charge may be too great, and this seems to be true whether this amount of fuel is inspirated through a jet or injected with a syringe. On the other hand, in spring and in fall weather, a 1-cc. charge may often be inadequate.

We seem to find that the amount of accelerating charge needed depends considerably upon the temperature of the air and the fuel as they enter the mixing-chamber of the carbureter; in other words, upon the instantaneous evaporation. More accelerating charge is of course needed when the hot-spot is a considerable distance from the carbureter than when it is fairly close to the jet.

The tests by Mr. Eisinger seem to show that the need of an accelerating charge is chiefly present at low engine-speeds. This is certainly true as regards eliminating the tendency of the engine actually to miss when the throttle is opened quickly. Numerous times, however, I have seemed to find that it takes just as great an accelerating charge to make the engine "bite in" and develop full torque instantaneously when the throttle is opened at a speed of 20 m.p.h. from a lean mixture as it does to give equally positive response at a speed of 5 m.p.h.

For about 2 years I have been driving different cars that were equipped with a manual control of the accelerating charge along with distant thermometers that indicate the temperatures of the hood and of the carbureter, and I have been impressed by the extent to which the "pep" and the response of the engine depend upon a proper accelerating-charge for the existing temperature. Nearly every driver has had the experience that, with any given carbureter-setting, the engine will first, after starting, be weak, as from too lean a mixture. Then, as the temperature under the hood rises, it will reach a condition of maximum response and flexibility. With still further driving, this maximum will be passed and the response to the throttle will become slower and less lively.

By proper coordination of the accelerating charge with the temperature, feeding more fuel at low temperatures and less fuel at high temperatures, the maximum responsiveness can be maintained. This is in marked contrast with the mixture requirements for steady running, which do not seem to change appreciably throughout the range of operating temperature to which I have just referred.

G. A. BURN[14]:—Heretofore the results obtained in acceleration runs have been very unsatisfactory. In comparing some of the results obtained in our laboratory with those of Mr. Eisinger's work, we find that, when using the engine of a popular light car and accelerating speeds from 400 to 1000 r.p.m., fuels of widely different volatility give approximately the same results. However, it seems that, had we taken a much lower speed from which to accelerate, differences could have been noticed.

Mr. Eisinger's work seems to indicate that, for maximum acceleration, the fuel volatility and the temperature of the manifold depend on one another. Theoretically, a relatively cool manifold and a volatile fuel seem to be the best combination, as these conditions would give a high volumetric-efficiency as well as good acceleration during the warming-up period of the engine. The design of the intake-manifold seems to be the main thing in acceleration which depends upon the distribution of the fuel-mixture. Probably some relation can be worked out between volatility, manifold temperature and velocity, somewhat similar perhaps to the relations discussed in the paper on Fuel Requirements for Engine Starting[15], by C. S. Cragoe and J. O. Eisinger.

[14] M.S.A.E.—Supervisor of engine laboratory, Tide Water Oil Co., Bayonne, N. J.

[15] See THE JOURNAL, March, 1927, p. 353.

# OPERATION AND MAINTENANCE

(*Concluded from p.* 116)

Varying requirements as to safety-doors for motor-coaches.

The fact that present State regulation in some cases discourages the use of six-wheel vehicles.

## TO ACT IN AN ADVISORY CAPACITY

Reference was made to the Uniform Vehicle Code of the National Conference on Street and Highway Safety, of which Secretary of Commerce Herbert Hoover is chairman, and it was felt that the Subcommittee might be guided by this code to avoid doing considerable research work. T. D. Pratt and R. E. Plimpton were designated as a subgroup to study the code from the point of view of the motor-truck and motor-coach operators respectively and to submit a report on the desirability of the individual statutes. Mr. Pratt further agreed to develop the regulations regarding the registration of engine numbers; and G. W. Reimann was asked to ascertain the extent to which the varying requirements are being enforced by State regulatory bodies.

The Subcommittee agreed that the Society as represented by the Operation and Maintenance Committee should act in an advisory capacity to the many other bodies which are attempting to bring about uniform and sounder legislation, and cooperate fully with the Motor-Vehicle Conference Committee, the National Automobile Chamber of Commerce, the American Automobile Association, and other organizations in their activities on this general subject.

The personnel of the Subcommittee is F. J. Scarr, of the Scarr Transportation Service, chairman; F. D. Howell, of the Motor Transit Co.; Adrian Hughes, Jr., of the United Railways & Electric Co. of Baltimore; J. F. McMahon, of the Yellow Taxi Corporation; H. V. Middleworth, of the Consolidated Gas Co. of New York; R. E. Plimpton, of *Bus Transportation;* T. D. Pratt, of the Motor Truck Association of America, Inc.; and G. W. Reimann, of the American Railway Express Co. Those members who attended the above-mentioned meeting were Messrs. Scarr, Plimpton, Pratt, and Reimann.

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# Teaching Production Principles to Engineering Students

*By* Myron A. Lee[1]

Annual Meeting Paper *Illustrated with* Charts

THE problem of teaching engineering in universities is complicated because (*a*) a large percentage of graduates enter fields of work that require knowledge of the fundamentals of production, organization, cost accounting, and human relations; (*b*) the volume of literature on these subjects is so great that suitable selection must be made; and (*c*) mechanical engineering embraces a very large number of branches with a great volume of information that must be imparted in each.

This condition led the faculty of the Sibley School of Mechanical Engineering to believe that its teaching should be confined mainly to fundamentals and that these should be impressed upon students by specific application to given problems. This principle has been followed in arranging the courses in industrial engineering, and the author gives an outline of the curriculum to make clear the present status of the management courses. Students are given the usual engineering courses during their first 3 years and in the third year are given an elementary lecture-course in industrial organization. Those electing the industrial-engineering option in their senior year are given a course in cost accounting, corporation and investment finance; a lecture course in industrial relations, and lecture, laboratory and drafting-room work in industrial engineering.

In this last course the principles taught in the lectures are paralleled by application of them to individual problems assigned to the students. Each is required to lay-out in the drafting-room plans for a factory; prepare a chart showing the number of machines of each type required for the given production of such units as, for instance, an automobile transmission; locate the machines to best advantage; prepare a chart of organization; devise forms for production-control; make time-studies, and in every way conduct all the operations of a theoretical plant.

Although many graduates engage eventually in management and production activities, the fields they enter are so diversified and the educational period is so brief that it is impossible and probably undesirable to concentrate on details of application of production principles to any one industry. Few students know definitely what specific industry they will enter after they have been graduated; hence, the policy of using drafting-room and laboratory problems as a means of enabling the student to grasp and retain general principles of production that are applicable to any industry seems to be a sound teaching-policy.

NOW that a large percentage of engineering graduates enter fields in which a knowledge of the fundamentals of production, organization, cost accounting, and human relations is essential, and a great volume of literature on these subjects has become available in recent years, the problem of the university faculty has become, not how to find material for a course of industrial engineering, but what shall be chosen from the enormous amount of material at hand.

This condition is complicated at the Sibley School of Mechanical Engineering by the fact that we must also graduate men who are mechanical engineers in the accepted sense, and by the tremendous increase in information which we should like to impart in each branch of engineering as exemplified by advances made in the development of alloy steels, high-pressure steam-engines, Diesel engines, turbines, autogenous welding, and the like.

Because of this condition it has become the feeling of the faculty that our teaching should be confined mainly to fundamentals and to impressing these upon the memory by specific application rather than to educating the student merely in the specific applications. This principle has been followed in the arrangement of the courses in industrial engineering, and a brief outline of the curriculum will make clear the present status of the management courses.

All students are given the usual engineering courses during their first 3 years, in which it is possible to give a large amount of instruction in mechanics and design because of the exacting entrance-requirements in mathematics. During the third year all students are given an elementary lecture-course in industrial organization, which treats of industrial history, fundamentals of division of labor, mass production, standardization, specialization, organization, and like subjects. In the senior year those electing the industrial-engineering option, in addition to heat-power, mechanical laboratory, and electrical theory and laboratory work, take a course in cost accounting given by the Department of Industrial Engineering, a course in corporation and investment finance, a lecture course in industrial relations, and a course in industrial engineering consisting of lectures and laboratory and drafting-room work. In this last course the principle of demonstrating fundamentals by their application to a specific problem is applied.

It is recognized, for instance, in teaching machine-design, that a lecture or even a recitation course alone cannot accomplish the results that can be attained in the drafting-room. If a student actually designs some piece of apparatus, such as a triplex pump, he acquires and, more important, he retains a knowledge of machine-design which probably he could obtain in no other way. Moreover, this knowledge is not limited in application merely to the apparatus he has designed, but is, to a large extent, fundamental and applies to the whole field. This is particularly true if the drafting-room course is correlated with a carefully prepared and well presented course of lectures which show the general application of the principles of the subject.

The present industrial engineering course was developed in an endeavor to apply this teaching principle. The

[1]Head of industrial engineering department, Cornell University, Ithaca, N. Y.

various subjects included in the course are discussed in lectures, and as each subject is studied the student makes a specific application to his own drafting-room problem. This drafting-room work consists of laying out a plant to produce some simple assembly and, as each subject, such as organization and production control, is discussed in lectures, the student parallels the lectures with definite problems in the drafting-room relating to his own plant.

Through the courtesy of the Brown-Lipe Gear Co., blueprints for one of its transmissions and operation sheets for each part were obtained, and each student makes floor plans of a factory to manufacture a definite number of these transmissions. All students can make use of the same data but, as each is designing a plant that differs in capacity from the others, each man has a different problem. One of these transmissions was mounted in the drafting-room so that all students could acquaint themselves with its design and operation, and each student was supplied with sheets detailing the operations on each part. Also, as an assistance toward a thorough understanding of the problem, manufacturers' catalogs of all machines used in the operations were made available. Each machine number, as given on an operation sheet, refers to a definite machine, and data, such as name of builder, work size of the machine, floor space occupied, rated horsepower, and cost, were supplied for each machine.

Although the data by which each student determines the number of machines required in his plant are based on one type of transmission, his plant is laid out as it would be for the manufacture of several types. The first step in his problem is to determine the number of machines of each kind required. This is accomplished by filling out a chart in which every part that is used in the transmission is listed by number and name down the left side and every machine used for producing the parts is listed by number and descriptive name across the top. From the operation times, as given on the operation sheets, the number of hours per month required for each machine for each part can be computed, and a summary of the figures in the vertical columns gives the total number of machine-hours per month required of each machine. The shop is assumed to run 210 hr. per month. If the machines are assumed to operate on the average 80 per cent of this time, then the average number of hours that a machine will operate per month is 168. Dividing the total of a column by this figure gives the required number of machines of the type specified at the head of the column. As each student is assigned a different number of transmissions per month as the capacity of his shop, he will determine upon different numbers of machines than the other students and will thus be working on a different problem.

The student next draws floor plans for a factory to house the machines. He may choose either a single-story or multiple-story building of modern type constructed in conformity with the requirements of the industrial code and labor law as they relate to stairways, sanitary conveniences and other details. He locates the machines for convenient flow of material and groups them to facilitate proper supervision.

### Organization Chart and Production-Control Forms

Having the floor plans and data regarding machines, it is possible for the student to attack a large variety of problems. The first one attempted is to make an organization chart for the management of the shop, as in Fig. 1, after the theory of organization has been discussed in lectures and explanations have been given of the development of functional, or line-and-staff, organization, as industries grew too large for line organization. In the discussion of organization and organization charts, the charts shown in Principles of Industrial Organization, by D. S. Kimball, and in Management's Handbook are very helpful. The student is examined on his knowledge of the duties and functions of each department shown in his chart to make sure he has studied the subject and expended thought upon his work, which is regarded as more essential than that he arrive at an ideal solution of his problem.

Mechanisms of control and the use of orders and returns are next studied. It is assumed that a customer's order is received calling for some parts which can be shipped from stock and some which have to be manufactured. This originates shipping orders, production orders, work orders, material issues, and other documents.

The method of handling the issuing of stores and keeping a balance-of-stores ledger is explained and the forms usually employed in industry are discussed in detail. The student then devises sample forms, such as a requisition, material issue, work order, shipping order, production order, and job time-card. He must also be able to trace the course of each of these forms through the shop and tell who fills-in each item and his authority or reason for doing so; also the information that is obtained from each form, by whom it is obtained, and what uses are made of it. It has been found that this problem gives the student a real knowledge of the method of using orders and returns for the control of production. He also gains a knowledge of good practice in the make-up of forms, and an opportunity is presented for interesting him in the subject of modern job-time recorders, cost-keeping methods and tabulating machines.

The case of the subject of production control is typical of the way in which the lectures cover the general field and specific application is made in the drafting-room. In the lectures it is shown how, in a small industry of intermittent type, the work of production control must be cared for; but, since variety is small and the number of orders few, routing, scheduling and dispatching are carried on by the superintendent and foreman without formal mechanisms and possibly without even recognizing these functions as such. As the business grows, these informal methods become inadequate and the principle of division of labor is applied, which results in time-study and a formal planning-department.

Explanation is made of how the functional departments handle stores by means of requisitions, stores issues, move orders, balance-of-stores ledgers, and the like; how the time-study and methods men determine the best methods of performing detail operations and the time which should be required, developing and providing in advance the necessary tools and fixtures; how the planning department routes the work, schedules it and dispatches it through the shop; and how route charts, schedule boards and dispatch stations are used.

As business increases still further, the problem becomes, not one of determining the sequence of jobs at a given machine, but the determining of the number of machines to operate continuously upon one operation. Routing is cared for in design of the building and location of the machines, while scheduling and dispatching are provided for by conveyors. The business is thus pictured as having grown to a continuous industry.

The student's own plant is regarded as of the intermittent-process type, and the student applies the principles of production control to his individual problem; he

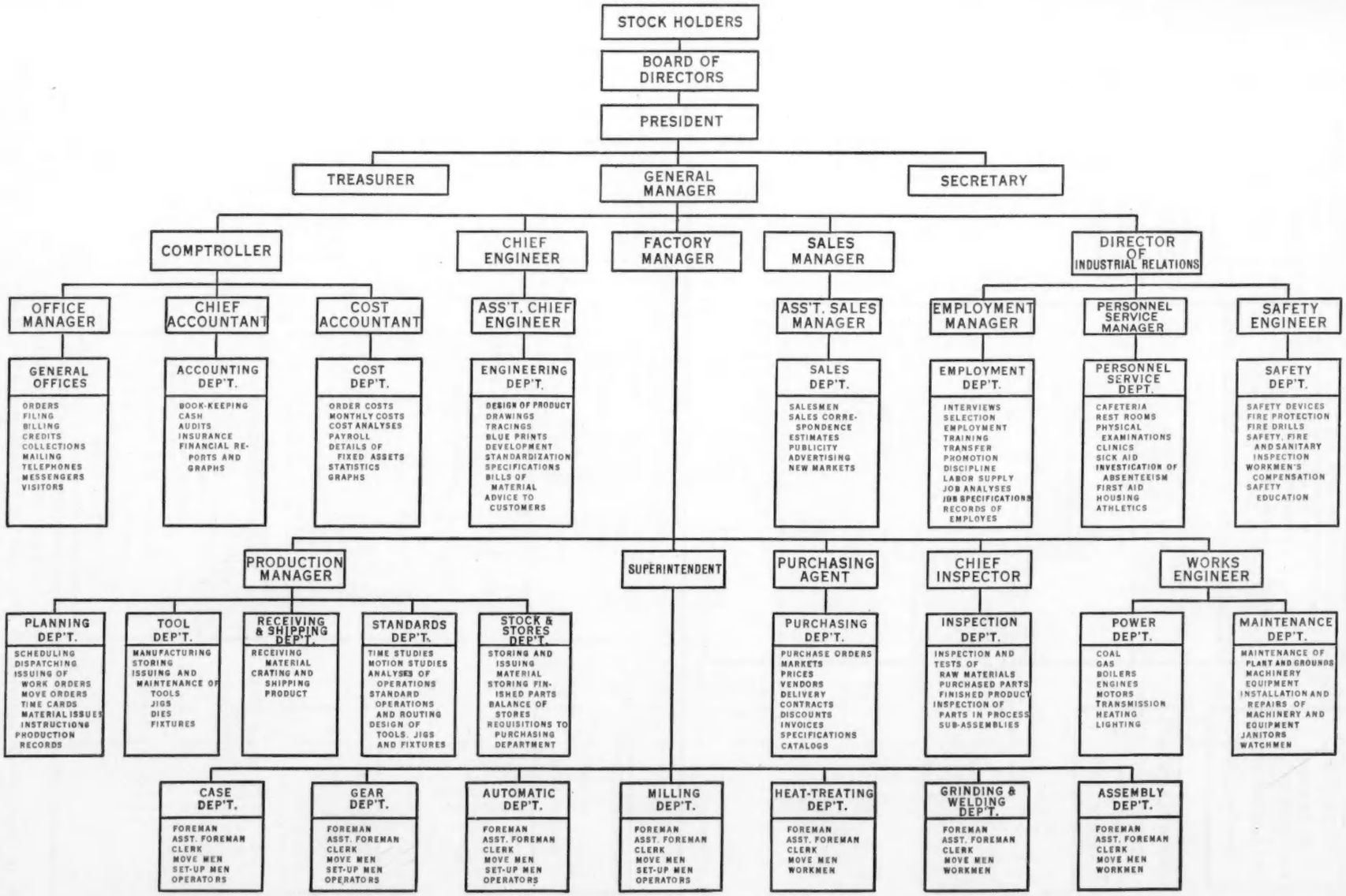


FIG. 1—TYPICAL ORGANIZATION CHART FOR A MANUFACTURING ENTERPRISE IN A MECHANICAL INDUSTRY

Each Student Taking the Industrial Engineering Course Is Required To Prepare a Chart of This Kind Applying Specifically to an Individual Problem That Is Given to Him. The Organization Must Be Suited to the Volume of Production and Other Conditions of a Plant Which He Lays Out in the Drafting-Room

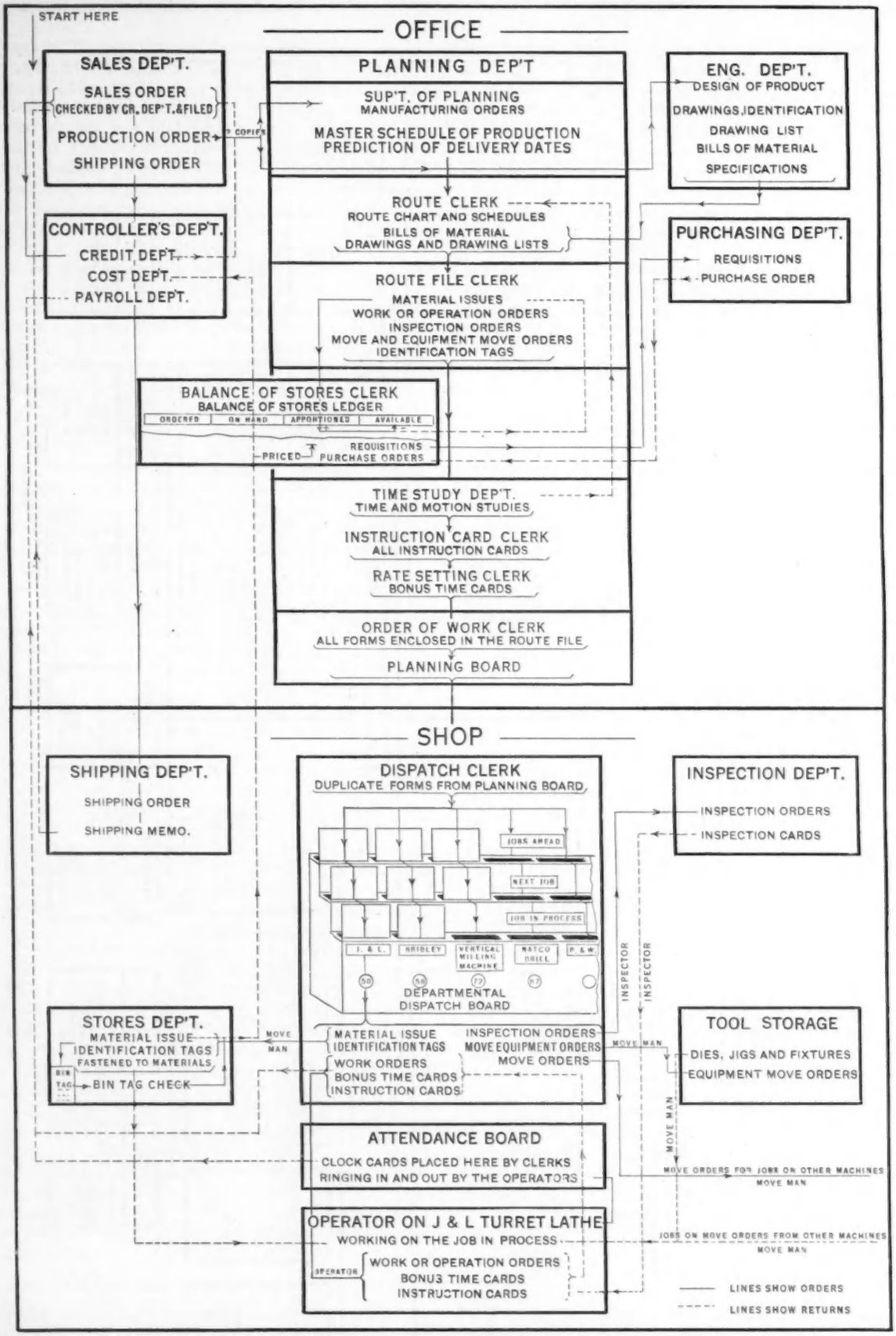


FIG. 2—CHART OF PROCEDURE IN A MANUFACTURING PLANT

This Is Used in the Lectures on Principles of Production Control to Show the System of Routing, Scheduling and Dispatching and Is of Great Assistance to the Student in Visualizing the Various Steps. The Solid Lines Show the Routes of Orders and the Dotted Lines Show the Return Routes of Requisitions, Reports, Memoranda, and Similar Documents

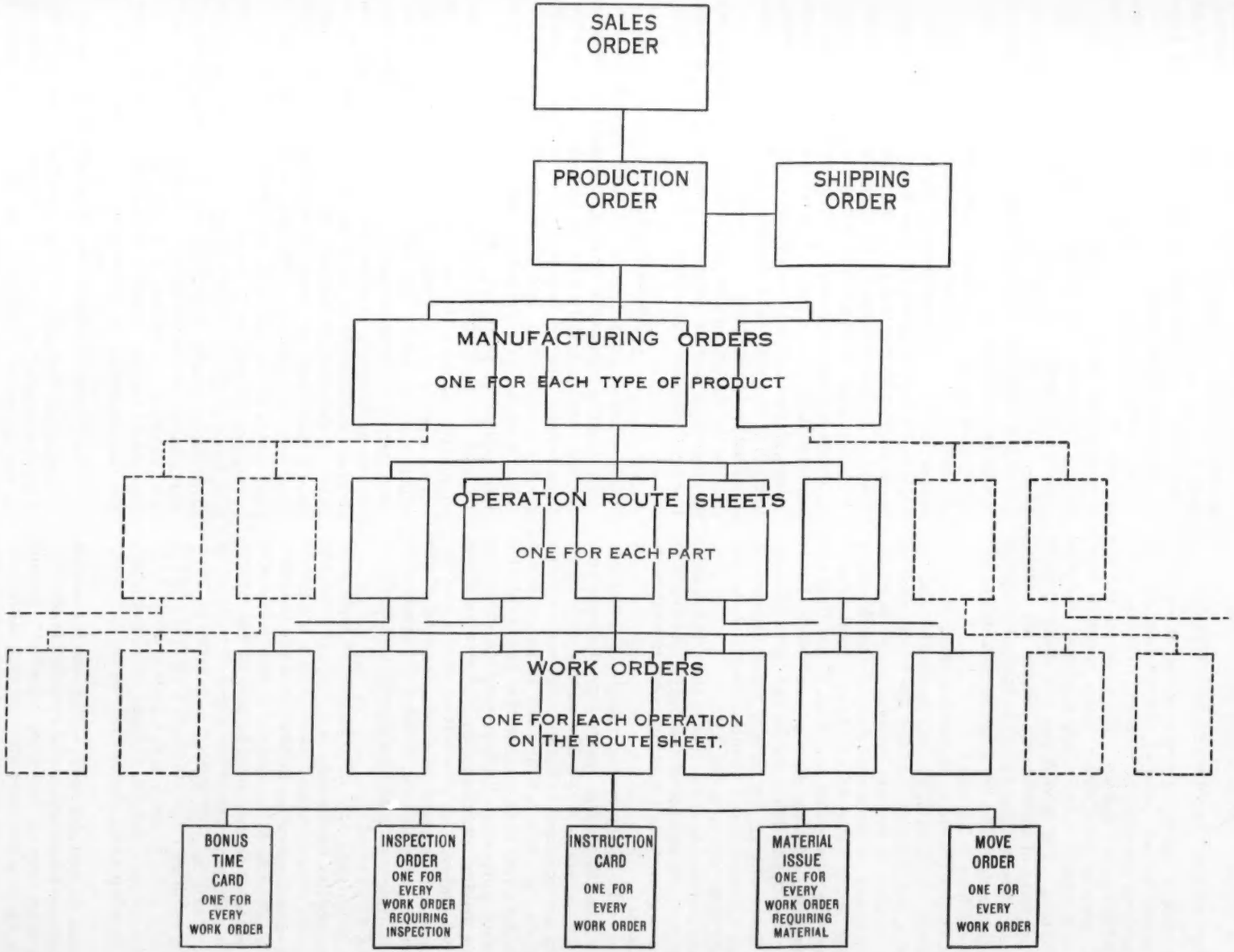


FIG. 3—CHART OF DEVELOPMENT OF ORDERS AND INSTRUCTIONS

This Chart Traces the Origin of Various Production Orders and Operation Instructions from the Sales Order through the Various Stages to the Move Order for Each Work-Order and Back to the Shipping Order. Dotted Lines to the Left and Right Indicate Provision for Expansion of the Number of Types of Product and Consequent Greater Number of Separate Operations. This Chart, Like That of Fig. 2, Is Also Used To Elucidate Lectures on the Principles of Production Control

devises the necessary forms and provides a system of routing, scheduling and dispatching. In the lectures, charts, such as those designed by Prof. J. R. Bangs and shown in Figs. 2 and 3, are of great assistance in aiding the student to visualize the necessary procedure.

### Application of Cost Accounting and Time-Study

After the student has had a course in cost accounting, in which he has already studied the relation of cost keeping to general accounts, he is ready to apply to his own plant the principles learned. The problems and application of process costs, order costs, standard cost, budgeting, and related subjects are explained and discussed in lectures. The advantages and disadvantages of the various ways of distributing expense are considered and then, in the drafting-room, the student computes an hourly expense charge by the machine-rate method for each type of machine in his plant.

In computing this charge he prepares a chart in which all the machines are listed at the left by number and descriptive name, and machine data and the hourly expense charges are entered in vertical columns under headings as follows: Number of Machines, Area of One Machine, Present Value, Scrap Value, Life in Years, Total Area of Group, Floor Area Chargeable to Group, Hours Used per Month, Horsepower Rating, Horsepower-Hours, Power Cost per Group, Depreciation for Group, Hour Charge for Group, Space Charge for Group, Value Charge for Group, Total Charges for 1 Month, Hourly Rate for Group, Hourly Rate, and Rounded Rate.

The charges are totaled at the foot of the cost columns. The student can now determine the manufacturing cost of a transmission made in his plant under normal operating-conditions.

Part of the work of the course consists in time-study practice in which the student makes time-studies of operations in the university shops. The method and objects of time-study are explained in the lectures, and emphasis is placed on the fact that a time-study is not merely a means of setting a rate for a piece-worker but is valuable for many other purposes. Various methods of taking readings and making allowances are discussed and the making of rate tables and time-setting charts are explained. Some time is spent in a consideration of the science of cutting metals and, insofar as practicable, methods of multiple-tooling and high-production machine-tool operation are studied.

As it has been found that allowing him to make preliminary time-studies results only in a waste of time, the student is given mimeographed sheets which enumerate the elements of each operation to be studied and the point where each reading is to be taken. Before taking the study he is required to demonstrate that he has practically visualized the sequence of the elements making up the operation, and he will almost automatically know when to take readings. The standard time-study watch, reading in 0.01 min., is used, and a person needs some practice before he can read one of these watches quickly, as he must do when using the continuous method of observation.

Each student is provided with a stop-watch mounted on a board of convenient shape and records his observations on the usual type of time-study log-sheet. Usually it is possible to have not more than 10 men taking observations upon one operator. Several complete operations or cycles are observed, to assure obtaining representative times.

After the observations are made, the student determines reasonable minimum times for all elements of the operation; makes the customary allowance for personal needs, usual delays and the like; prepares an instruction card for the operation, which determines the standard time for a given number of parts; and also writes a descriptive report of the study.

Each student makes a time-study of operations on perhaps five or six different jobs, such as operations on a molding-machine, a drop-hammer, drilling machines, and turret-lathes. For machining operations the cutting time is computed as well as observed, speeds, feeds, depth of cut, and other items being checked by tables, and the use of the Barth slide-rule is explained. This work does not make a student an experienced time-study man, and is not expected to do so, but it does familiarize him with the taking of a study and stimulates him to a much keener interest in the lectures on the subject.

Many of our graduates engage eventually in management and production activities, but the field they cover is wide. Because of this fact, and owing to lack of time, it is impossible and probably undesirable to concentrate on details of the application of production principles to any one industry. Since few students know definitely before graduation what specific industry they will enter, the policy of using drafting-room and laboratory problems, not to teach special applications but as a means of causing the student to grasp and retain general principles of production that are applicable to any industry, seems to be a sound teaching-policy.

## THE DISCUSSION

Chairman John Younger[2]:—I differ from Mr. Lee in one respect in training men; at Ohio State University we try not only to teach them fundamentals but to show them trends. We use text-books such as D. S. Kimball's for the fundamentals, and articles in *Automotive Industries*, *Manufacturing Industries*, *Factory*, and *Industrial Management* to point out the trends.

Lieut. W. S. Thee[3]:—To qualify a student for the automotive industry, the fundamental principles required can be given in a good course in mechanical engineering, which teaches mathematics, mechanics, electricity, chemistry, physics, thermodynamics, metallography, mechanical-laboratory work, machine design, machine-shop work, forging, pattern-making and industrial-plant construction. If the student desires to specialize in production, he can take optional courses in engineering administration, such as accounting, business management, industrial organization, statistics, and marketing methods.

Walker Gilmer[4]:—It seems to me that during the first few months of a man's time at college he should be given more insight into what can be accomplished, or what should be accomplished, possibly 10 or 15 years after his course is finished, and that the remainder of his time should be used in giving him a foundation of principles for accomplishing these results.

### Preparing Students for New Developments

Chairman Younger:—I look upon the situation of the student somewhat as in Fig. 4. Industry is progressing on the upward-sloping line. The student is at point *a* when he is graduated and it will be 5 or 7 years, more or less, before he reaches an executive position, as indicated by the intersection of the dotted and the solid line. We must try to give him all the background we can in school, but we also want to give him information about the new

[2]M. S. A. E.—Professor of industrial engineering, Ohio State University, Columbus, Ohio.
[3]S. M. S. A. E.—Instructor, Motor-Transport School, Quartermaster Corps, Camp Holabird, Baltimore.
[4]M. S. A. E.—Tacony, Philadelphia.

developments to come so that he will be able to fit himself to them when his time comes.

LIEUTENANT THEE:—Mr. Younger's thought is expressed by the saying, "The art always precedes the science." Science explains what happens, and the knowledge revealed by science often results in the development of new arts. Tetraethyl lead, for example, was discovered to be a suppressor of detonation of motor fuels. The scientist now has a problem to show how it does so, and in his work he may discover a better method of accomplishing the same results.

The college graduate who is trained successfully may lag temporarily upon completion of his college training but later will develop at a faster rate than the industry, as shown by the dotted line in Fig. 4. The key to success is the learning and development of methods of applying the fundamental and scientific principles that are taught in college. Success in any particular job or assignment in industry is attained by acquiring a knowledge and understanding of the fundamentals in the shortest possible time. The time required to do this depends upon the individual and the opportunities available within an organization in the way of an industrial training-program that is directly related to the industry.

The Army method of training its officers is to detail them to the General-Staff School, the Transportation School, the Administrative School, and so on. Engineering training should be elevated from the vocational plane to the professional plane. Professional training teaches fundamental principles and how to apply them; in vocational training the student merely memorizes established rules and one definite method of applying each.

PROF. MYRON A. LEE:—I agree fully that the only way to keep up with developments is to read the trade periodicals; there is no question about that. Lieutenant Thee's attitude on the subject is much the same as ours. When students come to the university we cannot know whether they will later enter the automotive industry or some other; therefore, the only thing we can do is to try to give them an engineering foundation which they can use in any industry.

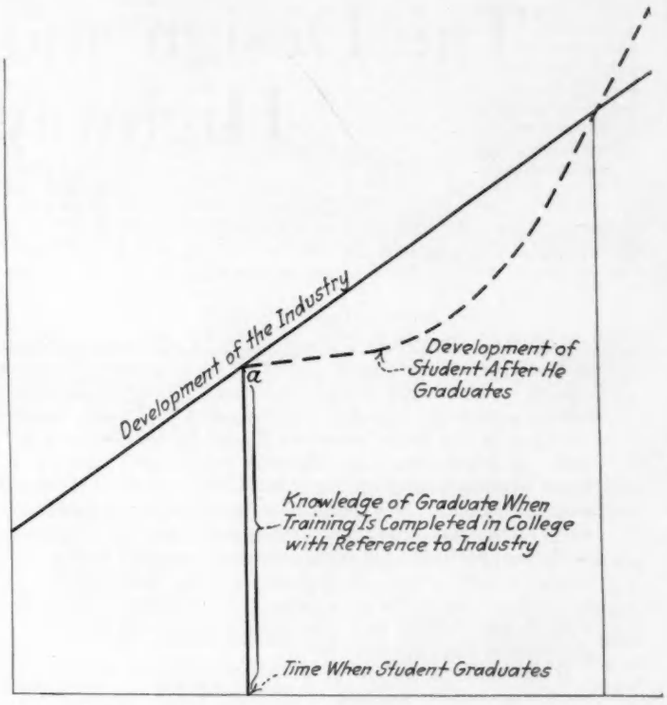
Development of the Industry
a
Development of Student After He Graduates
Knowledge of Graduate When Training Is Completed in College with Reference to Industry
Time When Student Graduates

FIG. 4—RELATIVE PROGRESS OF INDUSTRY AND THE COLLEGE GRADUATE
Industry Makes Progress as Indicated by the Solid Upwardly Sloping Line. When the Student Is Graduated After a 4-Year College Course He Is at the Point *a* with Reference to Industry and a Period of from 5 to 7 Years, as Indicated by the Dotted Line, Elapses Before He Becomes an Executive at the Time Indicated by the Point of Intersection of the Solid and Dotted Lines

# PRODUCTION ENGINEERING

(*Concluded from p.* 120)

they went through the same operations on the same machines and nothing apparently had been changed to improve the "noise" conditions. On the other hand, because of careless inspection, gears in which it is extremely easy to detect excessive noise are passed at times into the finished product only to be rejected. Again, one man will pass an engine, insisting that it is satisfactory, while an inspector will reject it on the next test.

In spite of the care taken in many cases to establish a standard to follow, considerable difficulty is experienced in obtaining a practical, uniform standard on noises. The securing of men whose sense of hearing is keen enough to enable them to "standardize" to a point where uniformity of product that will satisfy the buying public can be maintained is very difficult. According to our experience, about 1 man in 500 has a sense of hearing that will enable him to pass satisfactory judgment day after day; even the best men have their "off" days. Their hearing is affected by their physical condition and frequently by outside causes to the extent that they will pass unacceptable units which are not discovered until the final car-assembly has been completed. This is of course costly. It reveals clearly the weakness of the human element.

There is another difficult problem in standardizing inspection in matters where personal judgment is required, such as body finish, blemishes and off-color on painted bodies, and dirt, sags and scratches on enameled parts.

The invention of an instrument or process that will indicate or register noise in a practical manner would be one of the greatest possible assets in modern-day mass production. In visual inspection where personal judgment is required diligent effort must be made to pick men who are above the average in ability and have "even" dispositions; a temperamental or indifferent inspector cannot be educated to maintain a standard where personal judgment must be uniform from day to day to avoid a great handicap on production.

# The Design and Construction of Highway Systems

*By* THOMAS R. AGG[1]

METROPOLITAN SECTION PAPER *Illustrated with* PHOTOGRAPHS, CHARTS AND DRAWINGS

AN impetus to more scientific highway-construction was given by the World War, and, while it is necessary to limit the weight and size of motor-vehicles to safeguard the existing investment in highways, these limitations are being made as liberal as conditions permit. It is necessary for highway engineers to obtain a clear understanding of the interrelation of the highway and the vehicle. The author explains briefly the effects each has on the other and discusses these in connection with the problems of road location, grades, safety of users, cross-section of concrete slabs, the design of non-rigid road-surfaces, classes of highway and their cost to the public, and the economics of highway improvement and transportation.

It is believed to be possible, the author states, to show that, under certain conditions, road improvement creates wealth, either in the form of lowered transportation costs or as improved social and educational conditions, or both. Charts are given for the estimation of the approximate reduction in transportation costs and for balancing these against cost of road improvement and maintenance.

Highway engineers can reduce the cost of operating motor-vehicles by improving the highways, and automotive engineers can reduce the cost by further improvement of the vehicles.

The discussion deals with the kind of tremendously expensive highway improvements being made in New Jersey to carry the dense traffic and its economic aspect; the possibility of driving motor-vehicles by propeller and distributing the tractive effort through four wheels to reduce the burden and wear on the road surface; governing vehicles to operate at uniform assigned speeds on designated highways to reduce accidents; the best possible type of road construction for a heavily traveled trunk-line; and whether automobiles should come before good roads or the good roads before the vehicles.

THE World War, which brought about many changes in economic and social conditions in the United States, may be credited with having ushered in a new era in highway construction. While several years were required for the new ideals to gain general acceptance and an additional time for them to affect materially the construction practice of the highway departments, the impetus to more scientific highway-construction was due, in large measure, to a realization of the significance of war-time experiences with highway transportation.

For a long time after the motor-vehicle became the predominant factor in highway traffic, the attitude of highway officials was one of insistence that the vehicle must adapt itself to the highway. Now every well organized highway department is trying to adapt its highway system to the traffic. So far as finances permit, construction and maintenance programs are based on the adequate accommodation of present and prospective users of the highways.

The present investment in highways is so large and the need of additional mileage of surfaced highways is so urgent that there is now, and will continue to be, the necessity for limitations on loads and on size of vehicle that will safeguard the existing investment in highways. Regulations to this end are being made as liberal as conditions permit. In this phase of highway administration there has, on the whole, been a surprisingly cordial spirit of cooperation between the manufacturers and operators of vehicles and highway officials, notwithstanding some cloudy spots on the picture.

This new conception of the function of our highway systems has made it imperative that highway engineers obtain a clear understanding of the interrelation of the highway and the vehicle that uses it; of what the vehicle does to the road and what the road does to the vehicle.

The difficulty of securing accurate information on this subject has been tremendous because numerous technical and purely theoretical problems are presented which are too often unintelligible to those who control the purse-strings. Consequently, a pitifully small allocation of funds has been made for the investigation of what constitutes the best highway for each traffic situation.

In this paper an attempt will be made to point out

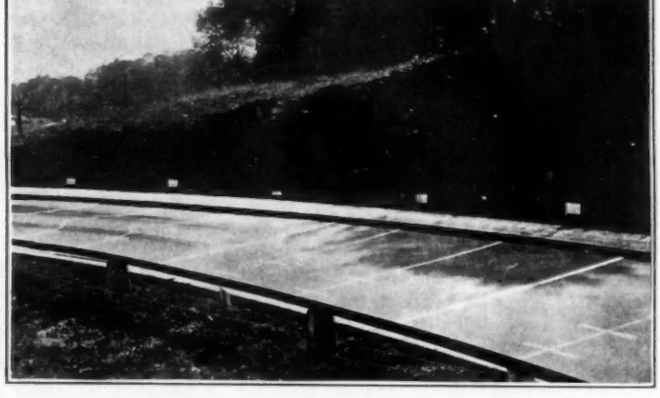

FIG. 1—DISPLACEMENT OF A NON-RIGID ROAD-SURFACE BY TRAFFIC
All the Transverse White Lines Were Straight Originally but Those at the Left Have Been Pushed into Various Wavy Forms by Creeping of the Road Surface Due to Action of Vehicle Wheels. This Is an Evidence of Improper Design or Construction of Roads That Are Somewhat Plastic and Is an Effect That Is Noticeable on All Intermediate and Low-Type Road-Surfaces. It Is a Factor That Must Be Taken into Account in the Designing of Roads That Carry Traffic in Excess of 300 or 400 Vehicles per Day. Photograph Supplied by the Bureau of Public Roads

[1] Professor of highway engineering, Iowa State College, Ames, Iowa.

some of the more recent developments in connection with the design and construction of highways and the economic basis upon which highway improvements are justifiable.

### What Vehicles Do to the Roads

Perhaps a start can be made by considering some of the things the vehicle does to the road:

First, the tires wear away the road surface. This is due to the tangential force applied by the driving wheels and the wheels that are fitted with brakes. This may be assumed as roughly proportional to the weight of the vehicle and the square of the speed. While this effect must be taken into account in connection with all types of road surface, it is especially important that it be evaluated correctly when the non-rigid types of surface are being designed. This is illustrated in Fig. 1. Bituminous surfaces are now being designed and constructed that have a toughness and stability undreamed of 10 years ago, which is being accomplished in part by a more scientific grading of the mineral aggregate and in part by a more careful control of the kind and quantity of bitumen used.

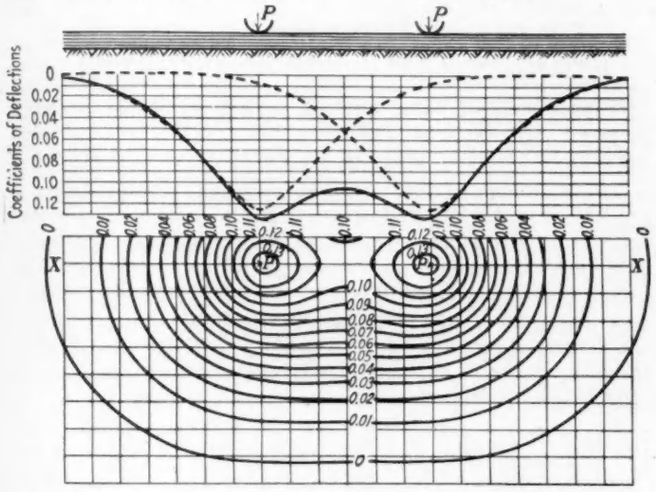


Fig. 2—Distortion Stresses in Concrete Slabs under Wheel Loads

Investigations Show That the Weight on a Wheel at the Center of a Slab Tends To Depress the Slab in the Form of a Saucer, as Indicated by the Contour Lines, and That the Effect Does Not Extend All the Way across the Slab or Road Surface to the Same Extent. Deflection Curves in the Upper Chart Are for the Section *XX* in the Lower Chart. The Dotted Lines Represent Stresses for Each Load Separately, and the Solid Lines the Stresses for Both Wheel Loads Combined. *PP* Are the Centers of Pressure

Loose "mulch" over gravel surfaces is a means of protecting this type of road from progressive wear. The use of bituminous surfaces on macadam and the more recent adoption of bituminous binders to minimize wear on gravel surfaces were steps in the direction of better protection of the road surface against the tangential force exerted by rubber tires.

Second, the road surface is distorted by wheel loads. If the road is of the rigid type, the distortion tends to depress the surface into a dish-like form, as illustrated by Fig. 2, the wheel load being in a position corresponding to the center of the dish. Distortion of this kind produces, not only simple tension and compression, but also torsional stresses of an exceedingly complex and elusive character. The Bates, Ill., experimental road, the Pittsburgh, Cal., experiments, and the stress measurements[2] that have been made by the Bureau of Public Roads represent an impressive series of studies of this subject. As a result, the engineer now has a scientific basis for estimating the strength required for rigid road-slabs and of rigid foundations for plastic surfaces.

[2] See *Public Roads*, April 26, 1926, p. 29.

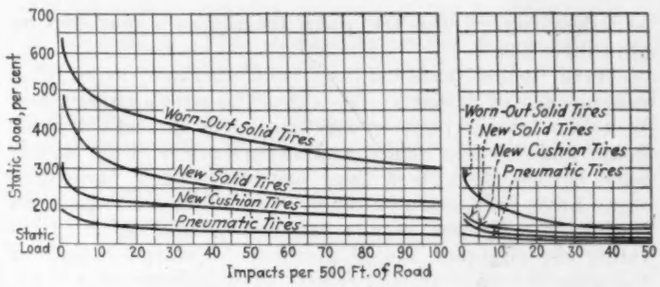


Fig. 3—Impact Loads on Road Slabs

Curves on the Chart at the Left Show Magnitude of Impacts on a Rough Stone-Block Pavement and Those in the Chart at the Right on a Smooth Concrete Pavement. In All Cases the Impacts Were Measured with a 2-Ton Truck, Fitted with the Rated Sizes of Dual Tires Loaded to Rated Tire-Capacity and Driven at a Speed of 12 M.P.H. Curves at the Right Correspond in Relative Position to Those on the Left for the Different Kinds of Tire

If the road surface is non-rigid, the distortion takes the form of a lateral and downward displacement. No formulas exist by which such surfaces can be designed; the engineer is compelled to rely upon experience and judgment, but it is certain that little hope can be entertained of securing stability with designs like those developed years ago for loads of 1 or 2 tons per wheel, although many highway boards seem determined to court disaster by so doing. Initial construction of inadequate road-crusts of gravel, sand-clay, macadam or bituminous macadam is not an irreparable error, because thickness can be added to these types without the loss of any considerable part of the value of that which is already in place. But if concrete or concrete-base pavements are too thin, strengthening is likely to be a very expensive operation and the complete salvage of the existing slab may be impossible.

Probably the greatest deficiency of present-day construction is the failure to provide a sufficient factor of safety in the initial construction of concrete bases and concrete roads. The standard sections adopted by the State highway departments too often fail to reflect a reasonable evaluation of the probable increase in loads during the life of the pavement, and the cities have shown

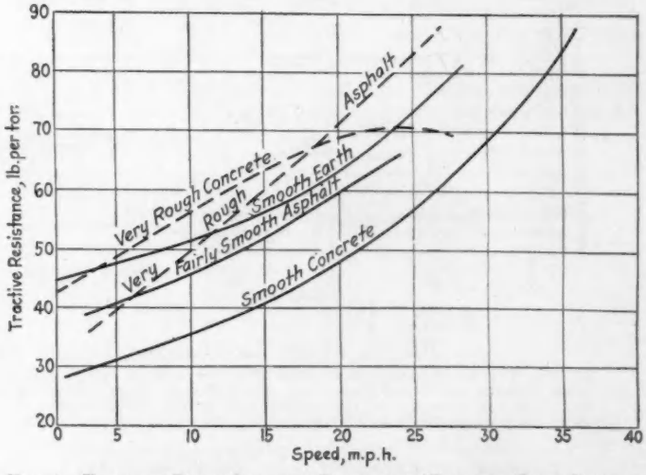


Fig. 4—Tractive Resistance on Different Kinds of Road Surface

The Curves Are for a Vehicle Weighing 1 Ton and Show the Rolling Resistance Plus Air Resistance at Different Speeds. Rigid Smooth Pavements Offer Little More Resistance than the Equivalent of the Power Losses in the Tires. Roughness and Yielding of the Surface Increase the Resistance, Which May Reach Several Hundred Pounds per Ton in Extreme Cases

still less regard for the plain lessons of experience in this direction.

Third, wheel loads subject road surfaces to varying magnitudes of impact. The extent to which this should be taken into account in any case is at present a moot point and further study and experience will be necessary before there is a final answer. Many investigations have been made to show the extent of impact on highways. Typical results are shown in Fig. 3. Certainly the engineer should give careful consideration to the possibilities of impact effects when he designs a roadway surface, but the exact allowance, if any, must be fixed by his good judgment.

### Effects of Road Surface on the Vehicle

The road surface has certain effects on the vehicle, and these should be evaluated in developing a highway system. It is possible in many instances to secure highway improvement without cost to the public by selecting the right kinds of surface for the several links in the highway system. This comes about through savings to the owners of motor-vehicles due to lowered operating-costs.

First, the road surface offers a resistance to the movement of the vehicle in the form of an interaction between the tires and the road surface. With high-type surfaces that are smooth, hard and unyielding this rolling resistance is but little greater than the equivalent of the power losses in the tires, or approximately 30 lb. per ton of weight on the tires. On low-type surfaces, the average resistance is considerably greater than for the high type and may reach several hundred pounds per ton in extreme cases, as shown by the typical tractive-resistance curves in Fig. 4. This resistance is a factor in vehicle operating-cost, because it is reflected in fuel consumption and tire wear.

Second, the road surface wears away the tire tread-rubber. This may be the determining factor in the life of the tire. It has been shown that the intermediate and low-type road-surfaces wear the tires at a rate of 4 to 10 times that of high-type surfaces and that in some cases the rate of wear on certain road-surfaces of loose angular stone may be at least 20 times as rapid as on a high-type surface. A few typical measurements of this effect are given in Table 1. So many variables enter into this sort of wear that it is impossible at present to point out just what influence any specific type of road may have, but the effect is always to be considered.

TABLE 1—APPROXIMATE RELATIVE WEAR OF TIRES ON VARIOUS ROAD SURFACES[3]

| Surface No. | Type | Tire Tread-Wear per 1000 Miles, Lb. | Relative Wear |
|---|---|---|---|
| 1 | High | 0.0662 | 1.00 |
| 2 | High | 0.1240 | 1.87 |
| 3 | Intermediate | 0.3520 | 5.30 |
| 4 | Low | 0.5600 | 8.50 |
| 5 | Very Poor Surface | 0.8630 | 13.00 |
| 6 | Abnormal | ..... | 17.00 |
| 7 | Sharp Stones, not Rolled | ..... | 56.00 |

[3] First and Second Progress Reports on Relation of Road Surface to Automobile Tire Wear, Engineering Bulletins Nos. 16 and 18, State College of Washington, Pullman, Wash.

Third, when a road surface receives impact from a vehicle, it naturally follows that the vehicle receives impact from the road. The resulting forces are mutually destructive to road and vehicle. Up to some critical speed, road roughness of a certain type increases rolling resistance and fuel consumption beyond that required for a smooth surface of the same type; at speeds greater than the critical, this effect decreases rather rapidly, as illustrated by Fig. 4.

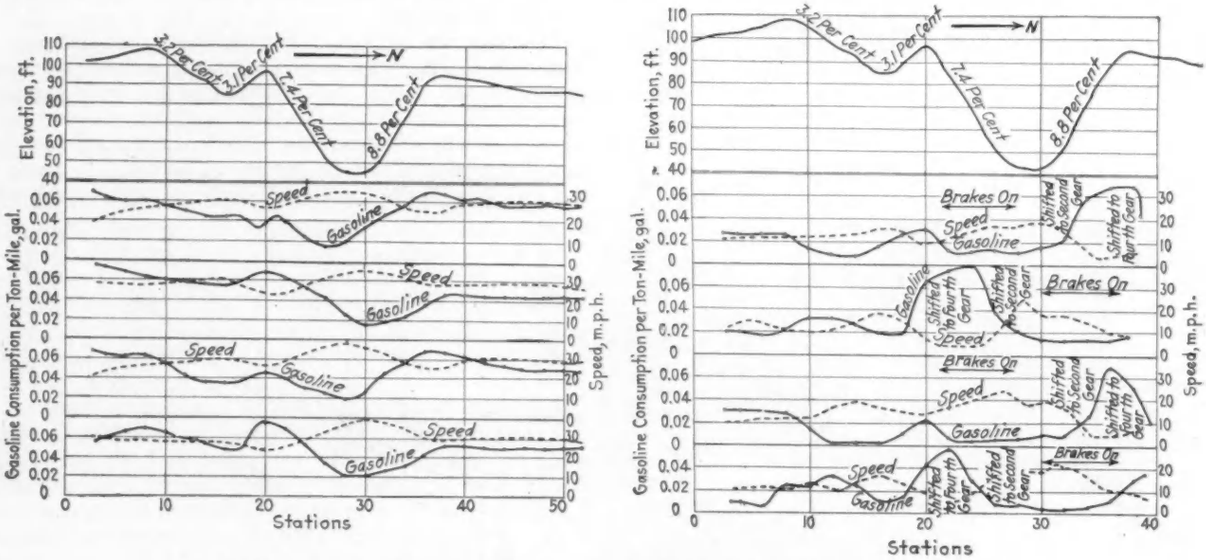


FIG. 5—EFFECT OF HIGHWAY GRADES ON FUEL CONSUMPTION BY MOTOR-VEHICLES

At the Top Is Shown the Profile of a Stretch of Hilly Road with Grades up to 8.8 Per Cent. It Extends from South, on the Left, to North, on the Right. A Buick Car Driven in Both Directions Showed the Following Gasoline Consumption:

| Direction | Clutch | Gasoline Consumption per Ton-Mile, Gal. |
|---|---|---|
| North | In | 0.0488 } |
| South | In | 0.0469 } Average 0.0479 |
| North | Out | 0.0475 } Average 0.0485 |
| South | Out | 0.0496 } 0.0 Grade 0.0480 |

For the Four Separate Runs the Curves Below, from the Top Down, Show the Relation between Speed and Gasoline Consumption on Each Part of the Road.

Over the Same Stretch of Road a Heavy Aviation-Truck Showed Gasoline Consumption as Follows:

| Direction | Clutch | Gasoline Consumption per Ton-Mile, Gal. |
|---|---|---|
| North | In | 0.0258 } |
| South | In | 0.0300 } Average 0.0279 |
| North | Out | 0.0184 } Average 0.0189 |
| South | Out | 0.0194 } 0.0 Grade 0.0220 |

Brakes Were Applied on the Steeper Grades, as Indicated, and Gearshifts Were Made at Beginning and End of These Grades, as Shown on the Records of the Individual Runs, Which also Show the Relation of Speed to Fuel Consumption. If the Grade of the Hill and the Condition at the Foot Make Coasting Safe, Considerable Saving in Fuel Is Possible

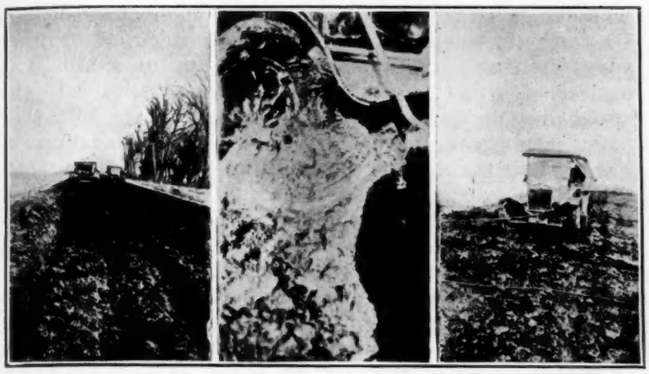

FIG. 6—KIND OF ROAD THAT IS COSTLY TO TRANSPORTATION
Part of the Lincoln Highway in Iowa in 1926. Mud and Dust, in Addition to Causing Time Loss, Increase Repair Costs and Depreciation on the Vehicle. These Effects Cannot Be Evaluated Accurately but a Good Guess Is that They Amount to an Average of between 1 and 2 Cents per Mile for Passenger Automobiles

## FACTORS AFFECTING ECONOMICAL-GRADE DETERMINATION

Fourth, the rate of grade of a highway may affect the time of travel and the fuel consumption of the vehicle. Many factors enter into the calculation of the most economical grade for any specific vehicle. Of these the significant ones are the maximum tractive-effort of the vehicle at the speeds considered and rolling and air resistance. The driving habits of the vehicle operator, permissible speeds and meterological conditions may have a bearing on the subject. It can be stated as a general principle that the most economical grade is one that will permit a vehicle to ascend in high gear at the most efficient engine-speed and to descend without the use of the brakes and without attaining too great a speed. As a general rule the grade that is just right for ascending vehicles is too high for descending; consequently, highway grades are always a compromise between the two. But it seems to be effectually proved that hilly roads are not uneconomical if a judicious selection of the maximum grade has been made. The curves in Fig. 5, selected from a large number of similar ones, illustrates this point.

Fifth, safety is becoming an increasingly important consideration in highway design. Slipperiness of the road surface frequently contributes to accidents. There seems to be but little difference in the coefficients of friction between tires and the accepted types of road surface when they are dry. When they are wet, the coefficient of friction may be greater than when dry, it may be about the same, or may be very much lower. This is shown by the data in Table 2. Likewise, a little frost on a surface may affect the slipperiness but little, or it may result in a very dangerous condition.

TABLE 2—TYPICAL COEFFICIENTS OF FRICTION

| Surface | Coefficient |
|---|---|
| Dry Concrete Road | 0.80[a] |
| Flooded Concrete Road | 0.94[a] |
| Dry Clean Asphalt | 0.86[a] |
| Water on Clean Asphalt | 0.90[a] |
| Wet Oiled Asphalt | 0.60[a] |
| Wood-Block Pavement, Dry | 0.53[b] |
| Wood-Block Pavement, Wet | 0.27[b] |
| Sleet, as Forming | 0.07[b] |
| Sleet, Roughened by Chains | 0.10[b] |

[a] Determination of coefficient of friction by Bureau of Standards; *Automotive Industries*, Nov. 29, 1924, p. 1095.
[b] Bulletin No. 67, Iowa Engineering Experiment Station, Ames, Iowa.

Sixth, dust and mud are factors in the deterioration of the vehicle. The exact cost of the depreciation and repairs chargeable to dust and mud has not been established. A good guess is that it lies between 1 and 2 cents per mile for automobiles. In Iowa we still have trunk-line roads that have to be navigated as in Fig. 6, which is from photographs taken on the Lincoln Highway in 1926. This kind of transportation is costly.

## MAJOR INFLUENCES ON ROAD LOCATION

These interrelations between the vehicle and the road are the factors that afford a starting point in the design of a highway, but the design that is dictated by the theoretical considerations involved must often be modified to meet such practical considerations as available funds, topography, cost of right-of-way, and especially the lethargy of public officials toward that which is new or abstruse. Some of the applications of the foregoing to rural-highway design are:

*Location.*—Three factors have the major influence on location. These are directness, grades and safety. It

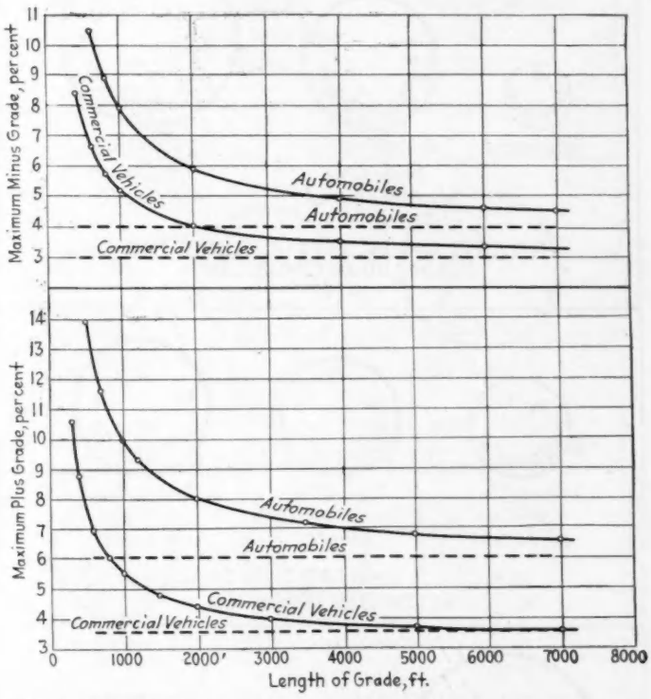


FIG. 7—METHOD OF DETERMINING RULING GRADES FOR HIGHWAYS
Positions of the Circles through Which the Curves Are Plotted Are Calculated by the Formulas

*Maximum Minus Grades*

For Automobiles
$P_m = 3.50/L(S_2^2 - S_1^2) + R/20$
$R = 80,\ S_1 = 30,\ S_2 = 45$
For Commercial Vehicles
$P_m = 3.65/L(S_2^2 - S_1^2) + R/20$
$R = 60,\ S_1 = 25,\ S_2 = 35$

*Maximum Plus Grades*

For Automobiles
$P_p = T/20 - R/20 + [3.50/L(S_2^2 - S_1^2)]$
$T = 200,\ R = 80,\ S_1 = 30,\ S_2 = 45$
For Commercial Vehicles
$P_p = T/20 - R/20 + [3.65/L(S_2^2 - S_1^2)]$
$T = 125,\ R = 60,\ S_1 = 25,\ S_2 = 35$

where
$L$ = Length of Grade, Feet
$P_m$ = Maximum Grade for Coasting without Use of Brakes, Per Cent
$P_p$ = Maximum Grade for Ascending Vehicle, Per Cent
$R$ = Tractive Resistance, Pounds per Ton
$S_1$ = Maximum Permissible Speed, Miles per Hour
$S_2$ = Maximum Desirable Speed, Miles per Hour
$T$ = Maximum Tractive Effort of Vehicle when in High Gear, Pounds

The Broken Lines Represent the Maximum Minus and Plus Grades for Automobiles and Commercial Vehicles at Which the Maximum Permissible Speed Equals the Minimum Desirable Speed

has been recognized generally that the shortening of a busy road is of great value because of the lowered cost of transportation thereon. Probably few persons realize the great possibilities in this direction. If a highway carries an average of 1000 tons of traffic per day, a total of 365,000 tons per annum, each foot of distance saved to the traffic will justify an expenditure of at least $60. For other amounts of traffic, proportionate savings would result, and these possible savings amount to stupendous sums on certain very busy roads.

Where the shortening of distance would entail heavy grading or the alternate of adverse grades, the designer must balance his design according to the exact conditions and in doing so should give due weight to the great value of distance saved, which might in many cases compensate for the cost of reducing adverse grades.

In determining grades, the first step is to decide whether it is permissible to utilize momentum effects. On tangents where there is no intersecting road or other speed-limiting condition near the bottom of a hill, it is entirely feasible to design momentum grades. If the speed must be controlled at the bottom of the hill, then momentum grades are inadvisable. The diagrams of Fig. 7 show the method of analyzing the grade situation. Grades in excess of those shown in the diagrams will often be employed and will result in lowering the minimum speed or raising the maximum speed, or both. It is possible, of course, to have a rate of grade that requires the shifting of gears and excessive use of the brake, in which case a certain extra cost for gasoline and tire wear will result, the magnitude of which can be estimated, although the analysis is too long for inclusion herein. This extra cost, however, may be much less than that of the extra distance required to secure lower rates of grade.

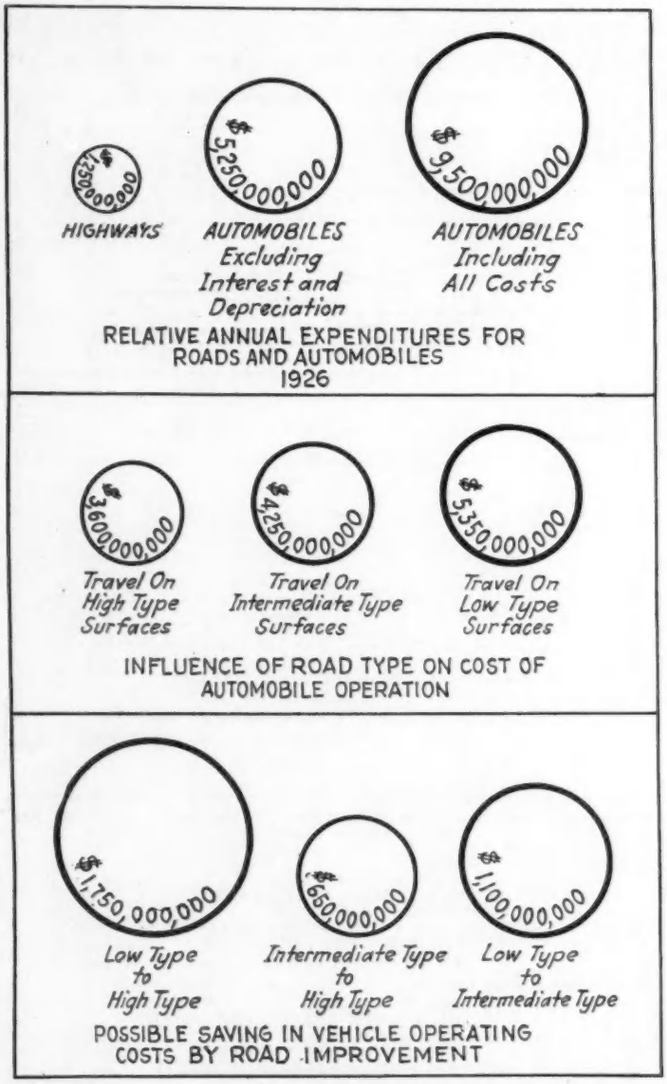
$ 1,250,000,000
$ 5,250,000,000
$ 9,500,000,000
HIGHWAYS
AUTOMOBILES Excluding Interest and Depreciation
AUTOMOBILES Including All Costs
RELATIVE ANNUAL EXPENDITURES FOR ROADS AND AUTOMOBILES 1926
$ 3,600,000,000
$ 4,250,000,000
$ 5,350,000,000
Travel On High Type Surfaces
Travel On Intermediate Type Surfaces
Travel On Low Type Surfaces
INFLUENCE OF ROAD TYPE ON COST OF AUTOMOBILE OPERATION
$ 1,750,000,000
$ 650,000,000
$ 1,100,000,000
Low Type to High Type
Intermediate Type to High Type
Low Type to Intermediate Type
POSSIBLE SAVING IN VEHICLE OPERATING COSTS BY ROAD IMPROVEMENT

FIG. 8—APPROXIMATE RELATIVE COSTS TO THE COUNTRY OF THREE CLASSES OF HIGHWAY

Actual Total Cost of a Road System to the Public Includes the Cost of Operating Vehicles on the Roads; That Is, All the Costs of Highway Transportation. The Upper Diagram Shows the Relative Expenditures in 1926 for Highways and for Automobiles, the Middle Diagram Shows How Much More It Costs To Operate on Intermediate and Low-Type Roads than on High-Type Roads, and the Bottom Diagram Shows the Possible Aggregate Savings That Can Be Effected by Raising the Low and Intermediate-Type Roads to Higher Types. Thus, a Saving of About $1,750,000,000 in Operating Costs Might Be Effected if All Low-Type Roads Were Raised to High Type

So far as available statistics are to be relied upon, the design of the road is but a minor factor in highway accidents. But short curves or blind curves, narrow embankments, narrow bridges, inadequate sight-distance on vertical and horizontal curves and at intersections of highways and of highways and railways all contribute to highway accidents. Those who use the highways agree that the elimination of grade crossings with railways should be prosecuted with the utmost diligence; there is less recognition of the almost equally urgent need for the elimination of grade crossings of the busier trunk-line highways.

Adequate superelevation of the roadway surface around curves is a factor that contributes to safety. This is most neatly and accurately accomplished when the tangent and curved sections are connected by an easement curve such as the cubic parabola that is employed in railway practice.

## THICK-EDGE AND REINFORCED CONCRETE-SLAB DESIGN

*Cross-Sections.*—Since the completion of the report on the Bates experimental road, there has been a widespread adoption of the thickened-edge type of cross-section for concrete roads and the use of embedded steel to carry shear at cracks and joints in the slab. Notwithstanding these marked improvements in design, the tendency has been to hold the thickness of slab to the minimum regardless of the soil and limatic conditions of service.

Two doubtful practices have crept into the design of cross-sections: One is to assume that a design complying with the epochal theory of Older with reference to thickness is sure to be adequate. The later analyses of Westergaard show that in some cases an inadequate factor of safety is provided on that basis. The other error is to assume that by designing concrete for high tensile-strength the slab thickness, and consequently the quantity of concrete, may be held to the minimum. Theoretically that is true, but evidence is accumulating to indicate that a certain element of safety against impact effects results from the inertia of the mass of concrete in the slab.

It seems that a logical basis for design would be to adopt for the concrete a mixture that would give no more than the necessary resistance to wear and to design the slab thickness on a basis of the modulus of rupture of the concrete thus obtained. There is also reason to feel that less difficulty will be encountered from temperature effects in the concrete meeting this requirement than with the richer concrete sometimes specified to secure a high modulus of rupture.

While the slab of uniform strength, of the thickened-edge type, has been widely adopted for concrete roads in preference to the slab of uniform thickness, this feature of design has gained adherents very slowly in connection with the design of the concrete base for sheet or block pavements. Yet there is ample reason to feel that the same principles apply. Moreover, the use of embedded steel for pavement bases of concrete has received but little consideration and this feature of design is believed to merit much wider adoption than it has received up to this time.

Another anomaly in the design of pavements is frequently encountered when concrete pavements are designed with embedded steel and transverse expansion-joints. Too often no provision is made to strengthen the pavement at the expansion joint and, when that is not done, the design is seriously unbalanced. So far as load-carrying capacity is concerned, the pavement is no better than one of plain concrete. The expansion joint may be strengthened by using dowel bars across the joint, with one end inserted in a tight-fitting sleeve that will permit the bar to slip, at the same time assuring that the slab will deflect alike on the two sides of the expansion sheet. The situation can be handled by thickening the pavement slab along the expansion joint just as it is thickened along the edge in other cases.

The design of the non-rigid roadway-surface requires the exercise of engineering judgment to an unusual degree, as no basis of calculation other than experience and precedent exists. Some road surfaces of this type are carrying loads equal to those imposed on rigid slabs and doing it economically. Many of these have been built up from thin slabs through years of maintenance until the requisite stability has been obtained. It is not safe to assume that similar designs will always be adequate for new construction, because a certain element of stability can be acquired only by long seasoning under traffic. These types are particularly adapted to programs of progressive improvement where it is expected that some failures will be encountered during the early life of the surface but that the weak places will gradually be eliminated by the maintenance forces.

### Maintenance Included in Road Cost

For convenience of discussion, the several kinds of roadway surface have been grouped, as is customary by many writers, in three classes: low type, intermediate type and high type. It is not possible to place each type of surface in only one of these groups, because some types, such as bituminous macadam, may be so constructed in one locality that they are logically classed as high type while in other places they are not so well built and are properly classed as intermediate type. Generally speaking, the well-known types of road surface fall into the three classes as follows:

*Low Type*—Natural-soil roads, sand-clay, light gravel, oiled earth, shale, shell, and similar materials.

*Intermediate Type*—Heavy gravel, surface-treated gravels and macadams, water-bound macadam and light bituminous macadam of various kinds.

*High Type*—Concrete, asphalt and block pavements with concrete base; heavy bituminous-macadams and the best of block-base types.

Actual unit costs for these three types are of little value in a discussion of this nature because these vary between wide limits according to the locality and the design. The significance lies in the relative costs of the three types, which also vary, but much less widely than the actual cost. The public is accustomed to think of price as synonymous with cost, which is entirely erroneous, as the two are often only remotely related. The true criterion of comparative economy is relative cost, in which the term cost is used in the broad economic sense.

The cost of any type of road is really the aggregate of the expenditure required to build that type of surface and keep it in a condition of maximum serviceability over an indefinite term of years, perpetually if so desired. This cost includes original cost and interest thereon, maintenance and interest thereon and depreciation, each of which factors must include a proper part of the cost of operation of the highway system, or overhead. When analyzed on this basis, the approximate relative costs of the three classes of highway for two traffic-densities are

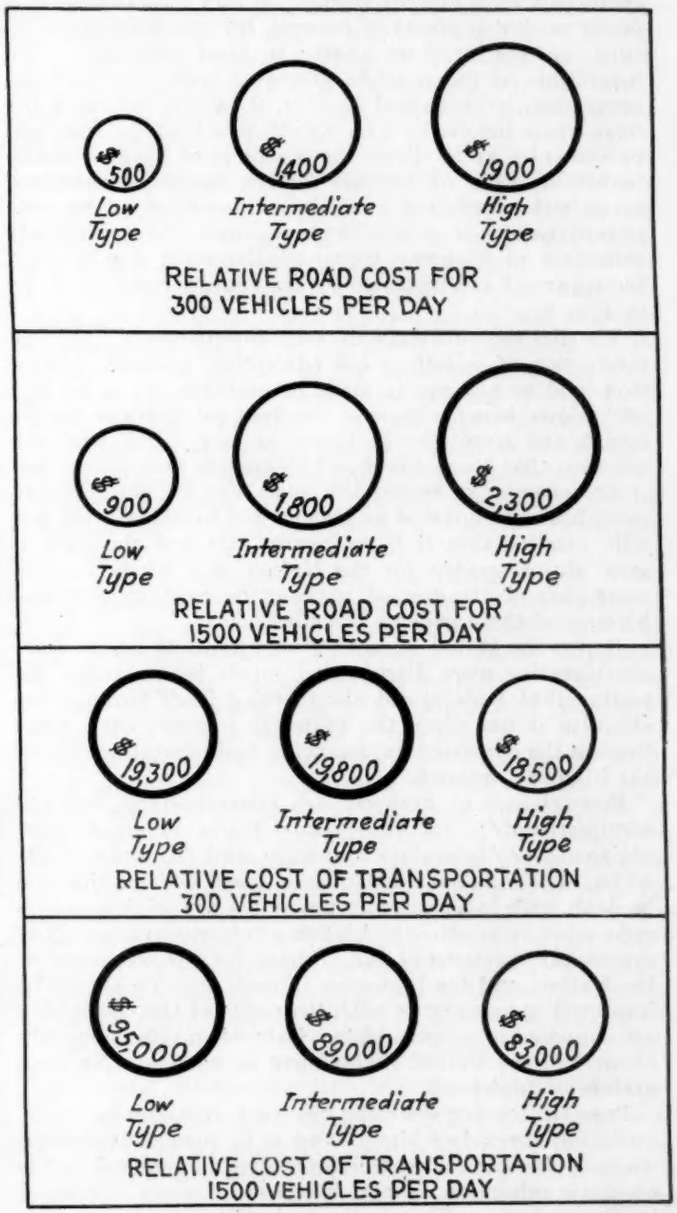


Fig. 9—Relative Cost of Roads and Transportation for Two Traffic-Densities

For a Traffic of 300 Vehicles per Day the High-Type Road Costs, in the Middle West, about $1,400 More than the Low-Type Road, Whereas the Cost of Transportation Is only $800 Less than on the Low Type. Therefore, the User and Taxpayer Saves Money by Maintaining the Low Type for This Traffic Density. But for a Traffic Density of 1500 Vehicles per Day, While the High-Type Road Still Costs $1,400 More than the Low-Type, the Cost of Operating the 1500 Vehicles Is $12,000 Less than on the Low Type; Hence It Pays To Improve It to the High Type

shown in Fig. 8. While these costs are admittedly approximate only, they show the general relation that exists. Exact comparisons that are applicable to a particular locality can be computed by using data from local experience. It seems that, in general, the lower-type roads are actually cheaper than the higher types for any traffic density under which the road could be maintained in good condition. The justification for road improvement must therefore be on some basis other than that of conserving road funds.

### Road Improvement Produces Wealth

It is believed to be possible to show that, under certain conditions, road improvement saves in transportation costs, that is, it creates wealth. If any road project produces wealth in excess of its cost, the necessary expenditure, no matter how great, is good business. The magnitude of the possible saving to traffic by road improvement is suggested by Fig. 9, which indicates the huge sums involved. The wealth produced by road improvement may be direct, in the form of lowered transportation costs, or indirect as the result of improved social and educational conditions brought about by road improvement. It is possible to estimate the approximate reduction in highway transportation-costs due to road improvement as illustrated by the general relations given in Fig. 8.

The indirect influence of road improvement upon the production of wealth is not susceptible of exact evaluation, and no attempt is made to evaluate the social and educational benefits because one does not appraise human health and happiness in terms of coin. But who will question that these direct and intangible benefits of road improvement far exceed the sums the Federal Government has appropriated as Federal aid to roads? Or who will question that it is eminently fair and the part of good statesmanship for the Nation as a whole to meet some part of the cost of nation-wide road-improvement because of these general benefits?

It may be stated parenthetically that, if every other consideration were disregarded, ample reason exists for feeling that Federal aid has justified itself through the stimulus it has given the States to improve their practices in the construction, financing and administration of the highway systems.

Four classes of highway are recognized by highway administrators: (*a*) purely local roads, (*b*) main market roads, (*c*) interstate highways, and (*d*) urban highways. These do not constitute separate systems that can be dealt with independently unless we are willing to concede some restriction to highway transportation. They are simply divisions of the arteries for the commerce of the Nation, and the highways in each division should be improved in accordance with the needs of the traffic they are supposed to accommodate. Only when this is brought about will the United States have an adequate National system of highways.

Possibilities for lowering the cost of highway transportation are of two kinds. One is by road improvement whereby the cost of vehicle operation is reduced to the economic minimum, insofar as those costs are influenced by the highway. This is the task of the highway engineer. The other is by improvements in the vehicle whereby its life is lengthened and the operating costs are as low as possible. That must be accomplished by the automotive engineer. Hence, the future trend of highway transportation depends largely upon these two great bodies of technical experts. Undoubtedly they will evolve roads and vehicles so well harmonized that the cost of highway transportation will be at the lowest possible rate. The benefits of their handiwork will flow out to every corner of the Nation.

### THE DISCUSSION

E. E. Reed[4]:—Professor Agg stated that the saving in cost of operation to the automobile user could justify an expenditure of $60 per foot of highway for every 1000 tons per day. I believe that is correct. I want to read something that Major W. G. Sloan, state highway engineer of New Jersey, has written in his report to the New Jersey State Highway Commission:

> It is obvious that, other things being equal, a saving in distance effects a saving in the cost of operation, and likewise a saving is effected by reduction in rise and fall and the elimination of grade crossings of railroads and important highways. As an example of the savings to be effected by these last-mentioned considerations, I will call your attention to the calculations made in connection with the extension of State Highway Route No. 1 to the vehicular tunnel, where it is estimated that the traffic to be carried will be approximately 18,000,000 vehicles per year. The capitalized value of the saving of 1 mile in distance under these conditions is more than $3,000,000. The capitalized value of the elimination of a drawbridge over an important stream is $8,000,000, and the capitalized value of the elimination of 20 ft. in rise and fall is $1,836,000.

At Professor Agg's rate in figuring the vehicles, the amount of traffic per year on this road extension would justify the spending of $37,000,000.

Our difficulty is to secure the necessary funds to build a comprehensive highway system, so we are not able yet to talk of paying $30,000,000 for reducing the length of a road 1 mile. The Highway Commission of New Jersey is proposing a highway from the Hudson River vehicular tunnel westerly to Elizabeth that will have no railroad grade-crossing and will not have an intersecting road at grade. You will realize what that will mean in getting to the South and to the New Jersey shore points.

### Road Work New Jersey Is Doing

The work that is now being conducted in New Jersey on the entrance to the vehicular tunnel is well worth investigation. The standard of construction on State highway work in New Jersey is this: We have a uniform slab of concrete 8 in. in thickness, minimum width of 20 ft. and with two lines of reinforcement. The roads are graded 40 ft. wide. The average cost of the concrete runs from $2.70 to $3.00 per sq. yd. Under ordinary conditions the average cost for 1 mile in the open country, where we have a pavement 20 ft. in width and graded 30 ft., is about $65,000.

The traffic counts we have taken show that the point of heaviest congestion is on State Highway Route No. 1 at Rahway, leading to the shore, where we had 43,735 vehicles in 24 hr. according to the count taken July 4, 1926. That was a peak load. The traffic at that point increased 64 per cent in 2 years. A 29-ft. pavement is carrying that traffic.

On the White Horse Pike, some sections built 7 years ago were 18 and 20 ft. in width and graded 30 ft. That road carries a traffic of from 20,000 to 30,000 vehicles some days during the summer. It leads from the Delaware River Bridge to Atlantic City. We found that this width was not sufficient to carry the traffic, and last year

[4] Assistant State highway engineer, New Jersey State Highway Department, Trenton, N. J.

that road was widened to 29 ft. by putting an additional slab on one side.

The approach to the vehicular tunnel is designed for five lanes of traffic and is 48½ ft. in width. Part of it is steel and concrete viaduct, and part is double-deck highway. It has to pass under some of the main parts of Jersey City, and that section, 3600 ft. long, will cost $4,500,000.

At the approaches to the Camden-Philadelphia Bridge there is an intersecting roadway or a crescent loop, which is 56 ft. in width. It skirts around the outside of Camden, intersecting all the roads leading into it, and takes the traffic to a central road 78 ft. in width that carries traffic to the bridge.

Those are the standards we have adopted on the State highway work, but, as Professor Agg intimated, we still have a vast mileage of township roads. In New Jersey the total road-mileage, including city streets, is about 20,000 miles. The State highway system comprises but 8 per cent of that mileage; 54 per cent comes under the jurisdiction of 233 different township committees, and most of that mileage is composed of gravel. The State aids to a small extent in the construction of such roads and approves gravel roads 20 ft. in width and bituminous-macadam penetration as narrow as 9 ft. in width and graded 20 ft. The idea in those cases is to get the farmers out of the mud. I have traveled over many miles of them and the 9 ft. is wide enough in some rural sections, because you can go some distance without meeting any traffic coming in the opposite direction.

### The Economic Stage of the Problem

E. P. Goodrich[5]:—Every problem goes through about four stages of development and study. The first stage is the one of lack of interest; we assume that things are going on about as they should. Then comes the stage when there is a recognition of the problem itself, but we usually study it in generalities. We make tentative comparisons and build on the basis of broad conclusions. The third stage is one in which we begin to analyze the several factors which enter into the problem. The fourth stage is the one in which each of the factors is analyzed to a conclusion, when we prescribe the remedy that will meet the particular phase of the disease which has to be worked out and eliminated.

This is an economic problem, fundamentally. Professor Agg showed in Fig. 9 the combination of the road plus the users' cost. It is that which the community and any large area or group of communities must study. It is that which Major Sloan is working on in New Jersey.

Of the various factors, there are two general divisions: those which pertain to the machine and those which relate to the road itself. In the machine, you men who are designing are looking for cheap first cost, because it is reflected in interest on the investment, in maintenance costs, and finally in depreciation. Then you are studying questions of gasoline and oil consumption. These are based fundamentally upon the friction factor which Professor Agg described, upon curvature to a certain extent and upon grade. It is those three factors in which the road engineer meets the automotive engineer. Tire and engine wear go into this same category in general.

Another item of cost involves speed, acceleration, braking, running time, and the time value of the machine as it is on its trip, the time value of the commodity handled or of the person or passenger who is riding it. This brings in the question of distance and of congestion. The traffic engineer enters the picture there and he studies questions of reduction of distance; increase in radius of curves; the elimination of those things, like grade crossings, which create congestion; and the method of operating so as to speed-up travel and reduce the stoppages from various causes.

The road costs and the first cost are large. Those types of road which have heavy maintenance costs and cheap first cost in all probability usually exceed annually the total of the interest on first cost together with the maintenance costs of the better classes.

There is a question of safety which is becoming more and more important and which is involved in the question of speed and of grade-crossing elimination. Automotive engineers can work with the road engineer by reducing the various factors of weight which produce friction and by reducing the costs of operation. The road man must react to produce a kind of surface which will give effective tractive and braking effort, with minimum wear on the pavement.

### How Motor-Vehicle Designers Might Help

Motor-vehicle designers have various opportunities for working to improve the situation from the point of view of the road engineer. If you could distribute the drive over all four wheels instead of placing it on two, you would get, with the same tractive effort, we will say, approximately one-half of the shearing resistance on the particles of the surface of the pavement, and you probably would reduce the costs of maintenance of the pavements at least one-half, because what produces the disintegration is that you have exceeded the elastic-limit or the total stress which will hold the particles of the pavement together. If you can get this distribution in drive, as you already have done in braking effects, you will benefit the whole situation.

If you can lighten the load by changing the whole drive system, using an airplane engine and propeller instead of the push of the rear wheels, you actually will be able to release some of the weight by taking it up into the air or lifting the load off the pavement. Therefore, it may be suggested that you change your scheme of things and branch out into a new field.

As to the element of safety, if you can operate all of your cars on any given stretch of road at the precise speed which is assigned for that road there will be no likelihood of one car overtaking another or stopping too suddenly for any reason, and accidents will be reduced. Therefore, install in your machines some governing device which will maintain a uniform speed when set at a specific rate, whether it be uphill or downhill or on a level. If you can operate uniformly, you will very largely cut out a certain class of accidents which now happen. These are just suggestions from the point of view of the road engineer.

### Good Roads Should Precede Automobiles

L. N. Whitcraft[6]:—Edward N. Hines, the father of Wayne County's concrete roads in Michigan, takes sharp issue with his good friend Henry Ford as to how to secure roads. Mr. Ford is said to have told a delegation of visitors at his Fordson plant that the formula for giving roads to the Country is to give the Country automobiles first. Mr. Hines does not agree with Mr. Ford's formula. Mr. Hines is known the Country over and also abroad as an authority on concrete highways. He built the first modern concrete highway in the world in Wayne County. His formula for securing roads is that roads come first. He says that automobiles are

[5] Consulting engineer, New York City.
[6] District engineer, Portland Cement Association, New York City.

found massed only where there are hard-surfaced roadways, and that they are few and far between where there are no improved roadways. He says, further, that the automotive industry would not be any larger than the pharmaceutical industry or of no greater import in the Detroit area than a dozen or so other industries if it were not for good roads. I am sure that all of you will agree with Mr. Hines. He also says the road must come first if there is to be maintenance of market in the automotive industry, and, if this industry is to write its own assurance of future growth, its leaders must take a far greater interest in every phase of improved-roadway extension and educational movements.

It looks to me as though everyone is either interested in automobile production or automobile use. No individual, group or industry is independent of its influence and benefits. Likewise, no one is independent of the influence and benefits of highways any more than he is independent of the influence and benefits of modern education. All go hand in hand and one is just as essential to our Nation's progress and development as the other. We continue to need more automobiles, more highways and more educational agencies. We need not only more highways but wider highways, safer highways and the expenditure of our highway funds for permanent construction. A great advance has been made in both the mileage and character of concrete-highway construction. To date, there are in excess of 45,000 miles of concrete highways throughout the Country. The highway engineers of the Country are building heavier and better highways than ever before. It is now generally acknowledged that it was a mistake to have built our earlier roads too narrow and too light. Recent surveys show rather conclusively that heavier roads are standing up well and meeting traffic requirements.

### Best Type of Heavy-Traffic Road

CHAIRMAN A. D. T. LIBBY[7]:—What do you believe is the best road it is possible to construct, Professor Agg? Suppose you were to be an engineer for the road between New York City and New Haven, which is heavy with traffic of all kinds, what kind of a road would you put in there?

PROFESSOR AGG:—I would put in a concrete road 12 in. thick, with about 40 lb. of steel per square, laid in lanes about 10 ft. wide, articulated so that I could add as many lanes as I cared to; and I would put expansion joints every 35 ft., with some kind of expansion-joint filler that would really take care of temperature variation.

JOSEPH A. ANGLADA:—What kind of foundation would you put under that road?

PROFESSOR AGG:—I should not care what was under it; I would use the best natural condition I could get and take care of the drainage. I long ago came to the conclusion that this question of good foundation for pavements is ridiculous; that no possible construction system can be devised that will keep water from seeping through the cracks in the pavement and softening the foundation to such an extent that you must depend for stability upon the inherent strength of the slab itself.

MR. GOODRICH:—In England last year I saw two samples of rubber pavement. They were built with slices of discarded automobile-tires placed in the joints between wooden blocks, with about ¼ in., perhaps, projecting up between the joints. This construction made a pavement which, so far as the experiment had lasted, was wearing less, was more quiet and gave better tractive effort than the wooden pavements which were common in the rest of the section.

CHAIRMAN LIBBY:—Was that kind of road good, in your opinion?

MR. GOODRICH:—I think it is along a new line which may work out. It is in the experimental stage at present.

CHAIRMAN LIBBY:—What is its cost?

MR. GOODRICH:—The cost, by using discarded tires and the saving in the wooden blocks, was just about the same as that of a wooden pavement, and it had many more advantages.

CHAIRMAN LIBBY:—If the wooden blocks are no better than those on High Street in Newark, it is a waste of time putting them down.

### Might Improve Use of Vehicles

R. E. PLIMPTON[8]:—I want to compliment Professor Agg not only on the thorough way in which he has summarized a complicated subject, but also for his frank and courageous discussion of certain "sore spots." Certainly these need attention, and plenty of it, if adequate remedies are to be provided. Those of us who are not in close touch with highway-development work may not know just how far-reaching these difficulties are. I should like to ask him to throw more light on the "cloudy spots" and tell us why they are, who is to blame and what the members of the Society can do, if anything, to improve the situation. He made an interesting reference to the outstanding deficiencies of present-day construction. I wonder if the State highway departments are entirely to blame for failure to look ahead and construct the concrete roads now being built with a sufficient factor of safety to take care of later increases in load. Is it possible that the practice is due to causes entirely beyond the control of these highway authorities?

Professor Agg referred to the lethargy of public officials toward that which is new or abstruse. Does he expect them to set practices in construction, which I assume he has in mind, if they are new and presumably unproved and untried? It would be interesting to know what methods there are which the public officials have failed to apply after they have been demonstrated as of practical value.

Lower costs of highway transportation certainly are desirable, but I am rather surprised that the author has limited the possibilities for accomplishing this decrease to two kinds: improving the highway and improving the vehicles. Both of these are important, but to many members of the Society a third possibility is still more important. I refer to the improvement of the use of the vehicle.

### Users' Highway-Cost Hidden

Many members who are in charge of operation or maintenance activities, in connection with large installations of motor-vehicles, are helping to improve the construction of their equipment and are continuously endeavoring to cut down operating costs by the application of approved methods and devices, and by getting together at such meetings as these for the discussion of their common problems. The interest of the Metropolitan Section in this subject has been sufficiently broad-gaged to lead to the suggestion that a National committee should be formed. The Committee on Operation and Maintenance was duly authorized by the Council of the Society and for the current year includes 24 members from a dozen different States. There are members on it in California, Colorado and Illinois, as well as a large number in the States in the Eastern part of the Country.

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[8] M.S.A.E.—Associate editor, *Bus Transportation*, New York City.

I take this opportunity to ask the cooperation of all the members in the activities of this Committee.

What does Professor Agg think about the possibility of the large operators establishing what is, in effect, a maintenance-of-way account, to correspond with the account carried under that heading by rail transportation-systems? Under the Interstate Commerce Commission practice the account is usually headed, "Maintenance of Way and Structures," and includes expense for stations and other buildings, as well as that involved in keeping up the track, bridges, culverts, and grading along the right of way.

Motor-vehicle operators are either taxed for the use of the highway or they are getting that use free. No one will admit that they are getting the use free, which means that the expense is being charged up under some other heading. It may be buried under a gasoline tax or licenses or some other form of taxation.

Professor Agg has been studying the economics of highway transportation for many years. He has figured out the total cost of operating motor-vehicles, including the charge for highway upkeep and construction, on different types of road surface. I am sure his opinion would be valuable as to just what the large operator should do, in fairness to himself and to the community, in studying the matter of highway costs that he is actually paying.

### Importance of Banking Neglected

L. F. Southwick[8]:—As highway construction is, in these days, a science in itself, and much time and energy have undoubtedly been spent in research work to improve the wearing qualities of pavements, smoothness of surface, non-skid surfaces, and what not, it seems strange that one feature which would at once improve motoring safety and comfort and save time has apparently been consistently overlooked. That is the superelevation of the outside of a curve, commonly called banking.

As the highways are improved, every possible means is taken to eliminate curves. It does not seem likely that it ever will be possible to eliminate them all, however. A properly banked curve will keep the vehicle on its own side of the road automatically, a result achieved only indifferently by the present method of painting a white line along the center of the road; it will avoid accidents caused by unfamiliarity with the road when the car is traveling at too rapid a rate; it will save the motorist's tires and brakes; and it will enable him to negotiate the turn safely without materially reducing speed.

It will be denied by none that average traveling speeds are increasing. This fact is appreciated by the State legislatures, because Connecticut, for example, set May 1, 1927, as the date on which a modification of the speed law would become effective to provide that the speed at which a man is driving will no longer be, as heretofore, prima-facie evidence of reckless driving. This attitude on the part of the legislature is undoubtedly in recognition of the trend on the part of drivers and also comes from a realization that, under present highway conditions, much higher speeds can safely be maintained.

The formula for banking is:

$$\text{Tan}\,\beta = 0.0668\,v^2/r$$

where

$\beta$ = the angle of banking
$r$ = the radius of the curve theoretically measured from the center of curvature to the center of gravity of the vehicle
$v$ = velocity of vehicle in miles per hour

This formula gives surprising results when the slight banking already given some curves is recalled. For example, suppose a vehicle is traveling around a curve of 500-ft. radius at a not unusual speed of 40 m.p.h. The banking angle, so that he will have no transverse reaction, will be about 12 deg. The natural tangent works out to 0.21376, or about a 2-ft. rise per 10-ft. horizontal width, for this speed and radius.

This formula, of course, cannot be applied indiscriminately, for the following reasons:

(1) The banking can never be so great as to endanger the equilibrium of heavily-loaded trucks, or trucks loaded so that the center of gravity is high. It should not be difficult to determine the limiting angle in such cases
(2) The foregoing applies only to level-road banking. Curves on hills will be discussed later
(3) Forty miles per hour is not necessarily an average speed, although it is coming to be such

The problem, then, consists of answering the questions:

(1) What speed shall be used in applying the formula?
(2) What is the limiting angle for safety for any vehicle at rest?
(3) Is it practicable or possible for contractors to construct curves banked much more than at present?

For the answer to question (1) we should look, not at today's speeds, but at those likely 5 or 10 years hence. The radius of curvature is an important consideration. Again, considering the radius, it seems that a speed higher than the average at present would be preferable from the banking standpoint.

It is only for comparatively short radii that the other important question, namely, the safe angle for all vehicles, would have to be considered. Assume a radius of only 100 ft. The angle in this case, for a speed of 40 m.p.h., would be nearly 47 deg. Obviously, such banking would be impracticable; but, on the other hand, such short radii are very unusual on the main highways.

Nevertheless, where the radius is sufficiently great that a speed of 40 m.p.h. would not lead to a dangerous angle, it seems preferable to construct the bank for this, or a higher, figure rather than for a lower one. We all are forced to lean toward the outside of a curve which is not banked or is banked only slightly. Why should we, so long as we do not topple over, mind being forced to lean toward the inside if we do not care to travel at 40 m.p.h.? Nothing is lost in either case, and much is to be gained for those who do prefer to travel at this, or a higher, speed.

### Banking Curves on Hills

Banking on hills is another important aspect of the question. Obviously, there are several more considerations involved here than for level-road banking. Whether the curve is at the bottom or the top of the hill, the severity of the grade, the extent of the curve, are all matters that have weight. They are to be considered because they have a bearing on the speed at which an ascending or a descending vehicle will travel.

To meet these conditions best, double banking is proposed. The banking at the top of a fairly steep grade for the up-coming vehicles would be very slight, while that for the down-coming vehicles could be figured as above. For the reverse condition, the curve at the

[8] Jun. S.A.E.—Sales service engineer, automotive department, Vacuum Oil Co., New York City.

bottom, the banking for both the ascending and descending vehicles would best be as great as practicable, with the same limitations as those outlined, although they would not necessarily be the same. In these cases it must be decided, from all considerations, what the most practical speed to consider will be, and the bank must be constructed accordingly. The road cross-section would then show two surfaces, at an angle greater or less than 180 deg. according to the position of the curve and its direction of curvature with relation to the grade.

It should be an extremely simple matter to construct from the formula a set of tables for the use of surveyors and contractors. In some cases a formula giving the sine of the angle, instead of its tangent, might be more useful. This would be used where the angle was large, and would obviate the necessity for figuring the horizontal projection of the road width. In other words, the tangent formula gives the superelevation per unit width of horizontal, while a sine formula would give the superelevation per unit width of road surface at the required angle. The sine formula works out to be:

$$\text{Sin}\,\beta = 0.0668\,v^2/(r^2 + 0.0045\,v^4)^{1/2}$$

and tables could be constructed from this almost as readily as from the tangent formula.

So far as the difficulty of constructing banks goes, it is doubtful if banking angles would ever be great enough to merit serious consideration on this score. Probably any degree of banking that would not prove dangerous to a standing vehicle would be easy to attain, if not with concrete, with a hot-mix to build it up to the required angle.

That suggests the last point, namely, that scientific banking is comparatively easy to incorporate in our present roads. In many cases a fill would do the trick, the fill to be either hot-mix or concrete. In some cases, particularly on hills, cutting would also be necessary, and in this case it would be better to wait until the road is to be reconstructed. In any event, it is true that today scientific banking is not practised when curves are laid out, and equally true that no one thing would tend toward greater safety, comfort and expedition of the motorist of today and of the future.

# JOHN RICHARD HALL

AFTER an illness of 7 weeks' duration, John Richard Hall passed away on June 11 at his home in Cleveland. His death caused not only grief to his wife, son and married daughter, but deep regret on the part of his associates in the Chandler-Cleveland Motors Corporation, of which he was vice-president in charge of production, and to his many friends in the automotive industry, with which he had been identified since its inception.

Born in Dublin, Ireland, in November, 1866, Mr. Hall came to this Country when 14½ years of age. He gained technical experience in the general machinists' trade, made a specialty of screw-machine work for several years, and was foreman of such work for the Metric Metal Co. at Beaver Falls, Pa. He was connected with the Lozier Mfg. Co. and Lozier Motor Co. for 21 years, during the first 13 of which he was foreman, and for 8 years thereafter, manager of the service department. In 1913 Mr. Hall joined the Chandler Motor Car Co. in Cleveland as factory superintendent. During his service with the Lozier Company he was sent abroad in a mechanical capacity and spent 1 year in England and 8 years in Germany, with headquarters at Hamburg. His conscientious institution of systematic shop-practices for the Chandler Company was rewarded by his election to the vice-presidency, and when the Cleveland Automobile Co. was consolidated with the Chandler Motor Car Co. he continued as vice-president in charge of production.

Mr. Hall was elected to the grade of Member of the Society in March, 1916, and was also a member of the American Society of Steel Treaters. He contributed throughout his productive life to the advancement of the automotive industry.

# ARMY MOTOR-VEHICLE DEPOTS RETAINED

FUNDS appropriated for the maintenance of the Army for the fiscal year 1928, which began July 1, made provision for the present mechanical and personnel set-up of the four major repair-centers of the Army. In making the allotment of funds to the big motor-vehicle depots, the War Department approved a continuance of the repair shops that on July 1, 1926, had been placed on probation for 1 year.

A year ago the major motor-vehicle repair-shops of the Quartermaster Corps at Camp Holabird, Md.; Camp Normoyle, Tex.; Fort Mason, Cal.; and Jeffersonville, Ind., were scheduled to be abandoned, it being proposed to maintain the Army's motor transportation by letting out the repair work to commercial shops. Major-Gen. William Hart, who was then Quartermaster General of the Army, believed that there was economy in the new plan. In November, 1925, Major-Gen. B. F. Cheatham became Quartermaster General and through his efforts the closing order was held in abeyance and the shops placed on probation to demonstrate that they could be operated economically.

The year of probation saw practically all civilian mechanics discharged and the work of maintaining the Army's transportation turned over to enlisted mechanics almost exclusively. Many short-cuts were made and savings in overhead and material costs began to appear during the first month. A new unit-replacement system was substituted for vehicle repairs. It took less than 3 months for the soldier mechanics to "catch their stride," and the production reports and cost sheets improved each month.

The morale of the soldier mechanics was never higher than during the last year. It was almost a foregone conclusion long before the year ended that the shops had demonstrated beyond question that they were an indispensable asset to the military establishment and would be continued indefinitely.—*U. S. Army Speedometer.*